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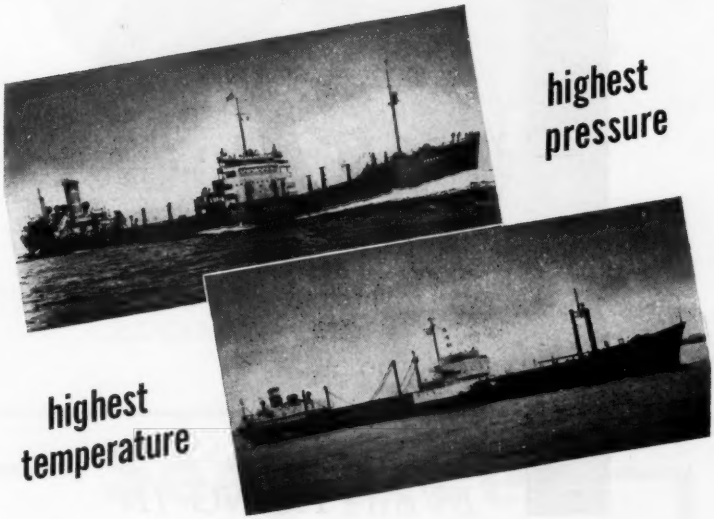

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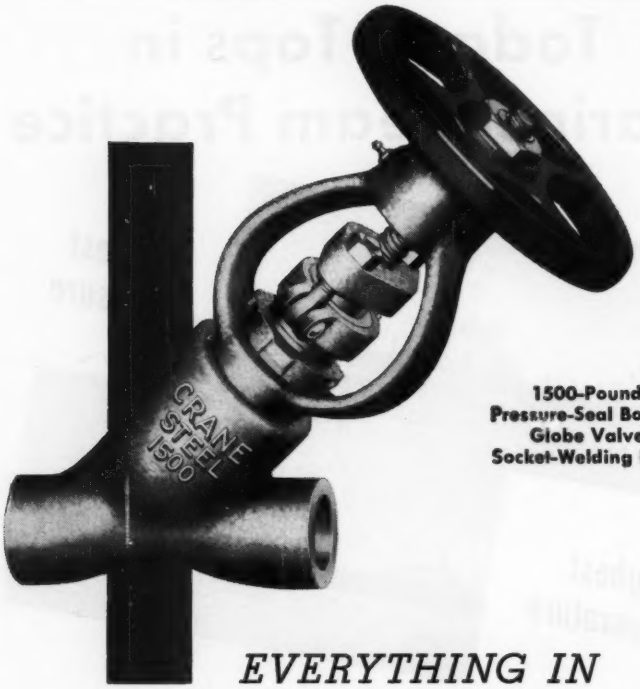

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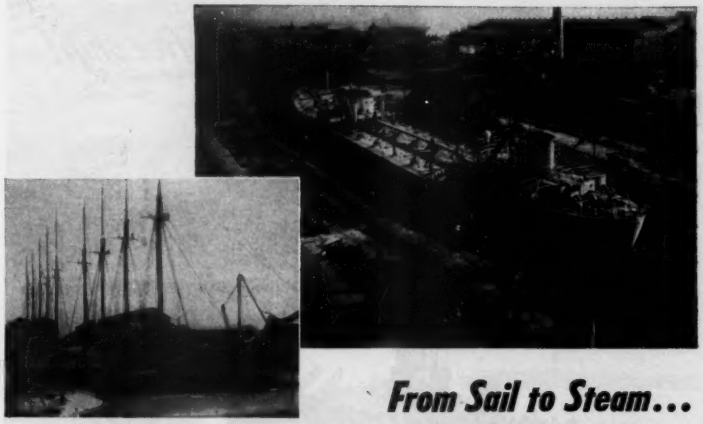

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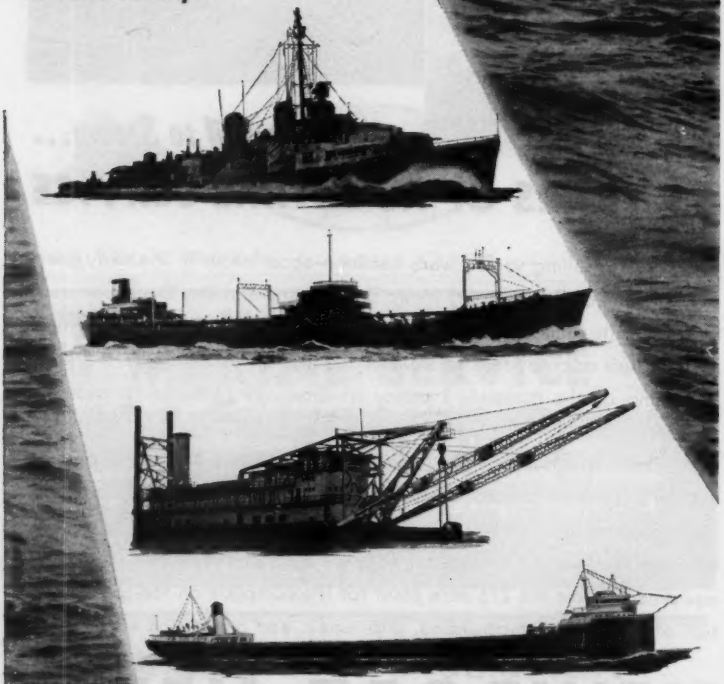

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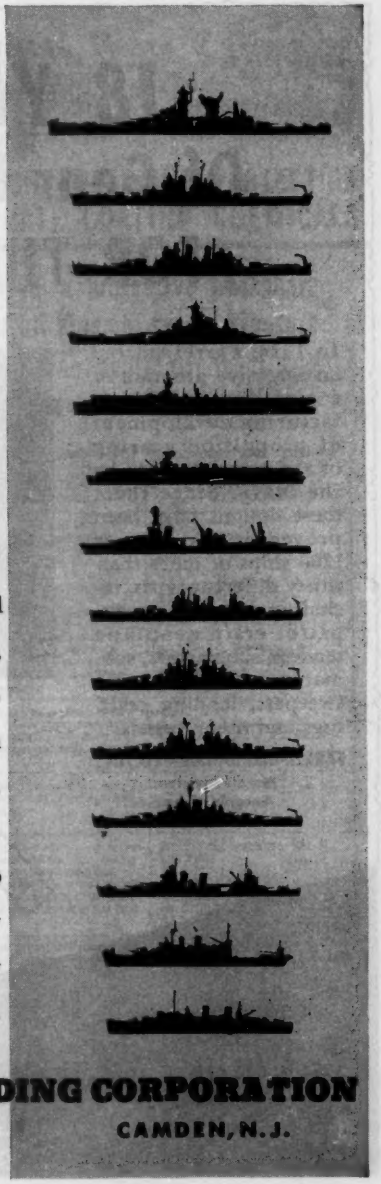

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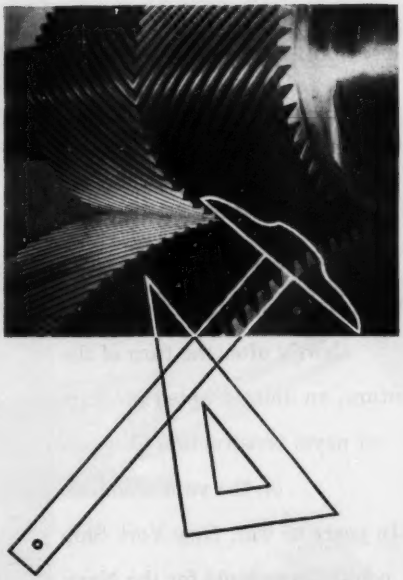

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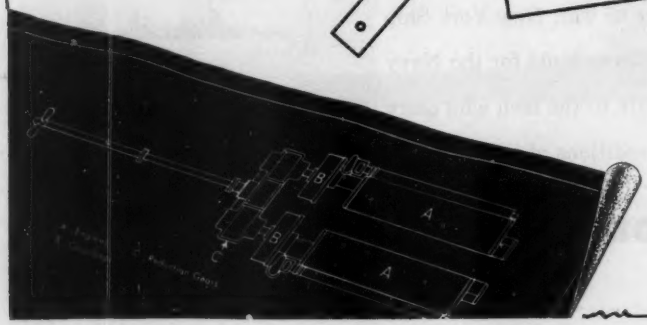

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STEEL	"GRANODINE" Zinc Phosphate Coating Chemical	Improved paint adhesion	Steel, iron, or zinc fabricated units or components, automobile bodies, refrigerators, washing machines, cabinets, etc.; projectiles, rockets, bombs, rifles, small arms, belt links, cartridge tanks, vehicular sheet metal, tank bolts and links, recoilless guns, etc.	MIL-S-5002 JAN-C-490, Grade 1 JAN-F-495 U.S.A. 57-0-2, Type II, Class C U.S.A. 51-70-1, Finish 22.02, Class C U.S.A. 50-68-1 16 E4 (Ships)
	"PERMADINE" Zinc Phosphate Coating Chemical	Rust and corrosion prevention	Nuts, bolts, screws, hardware items, tools, guns, cartridge clips, fire control instruments, metallic belt links, steel aircraft parts, certain steel projectiles and many other components.	MIL-C-16232 U.S.A. 57-0-2, Type II, Class B U.S.A. 51-70-1, Finish 22.02, Class B Navy Aeronautical M-364 U.S.A. 72-53 (See An-F-20)
	"THERMOIL-GRANODINE" Manganese-iron Phosphate Coating	Wear-resistance anti-galling, safe break-in of friction or rubbing parts. Rust proofing.	Friction surfaces such as pistons, piston rings, gears, cylinder liners, camshafts, tappets, crankshafts, rocker arms, etc. Small arms, weapon components. Hardware items, etc.	MIL-C-16232 U.S.A. 57-0-2, Type II, Class A U.S.A. 51-70-1, Finish 22.02 Class A Navy Aeronautical M-364 U.S.A. 72-53 (See AN-F-20)
	"GRANODRAW" Zinc-iron Phosphate Coating	Improved drawing, extrusion, and cold forming	Blanks and shells for cold forming, heavy stampings; tube; tubing for forming or drawing; wire; rod; etc.	———
ALUMINUM	"ALODINE" Protective Coating	Improved paint adhesion and corrosion resistance	Aluminum products of similar design such as refrigerator parts, wall tile, signs, washing machine tubs, etc; aircraft and aircraft parts; bazookas (rocket launchers), helmets, belt buckles, clothes dryers, clothesline, rocket motors, etc., aluminum strip or sheet stock.	MIL-C-5541 (See also QPL-5541-1) MIL-S-5002 AN-F-20 U.S. Naveord O.S. 675 16 E4 (Ships) AN-C-170 (See MIL-C-5541) U.S.A. 72-53 (See AN-F-20)
ZINC	"LITHOFORM" Zinc Phosphate Coating Chemical	Improved paint adhesion	Zinc alloy die castings; zinc or cadmium plated sheet or components; hot dip galvanized stock; galvanneal; signs; siding; roofing; galvanized truck bodies; etc.	QQ-P-416 RR-C-82 JAN-F-495 AN-F-20 U.S.N. Appendix 6 U.S.A. 72-53 (See AN-F-20)

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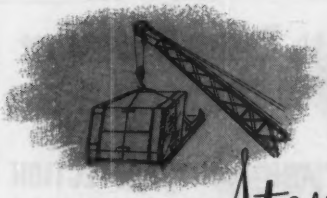
CHEMICALS ACP PROCESSES



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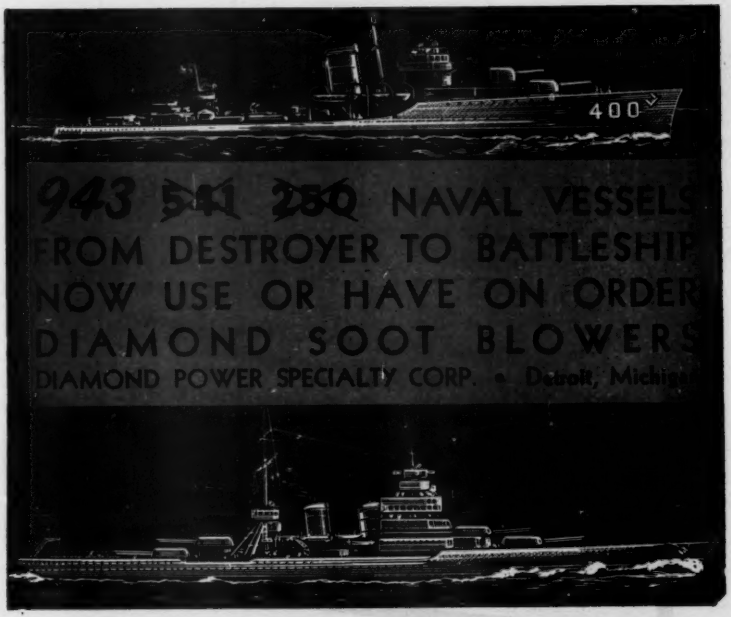

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REAR ADMIRAL T. C. LONNQUEST, U. S. N.
President of the Society
1952

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(Under whose supervision this number is published).

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Captain FRED WIESNER, U. S. N. R.

Captain J. E. HAMILTON, U.S.N., (Ret.), Secretary-Treasurer

SECRETARY'S NOTES

THE ANNUAL ELECTION

The ballots submitted in the annual election for officers and members of the Council of the Society resulted in the election of the following:

The President for 1952

REAR ADMIRAL T. C. LONNQUEST, U.S.N., Deputy Chief of the Bureau of Aeronautics, Navy Department, and Member, National Advisory Committee for Aeronautics.

Admiral Lonnquest was born in Lynn, Massachusetts, and was an honor graduate from Dartmouth College, including election to Phi Beta Kappa. He served in World War I as a Lieutenant in the old Naval Reserve Flying Corps, transferring at the end of the War to the Regular Navy. He is one of the few senior naval officers to hold a naval aviator's designation in both conventional airplanes and lighter-than-air-craft.

Admiral Lonnquest, in 1923, graduated from the U. S. Naval Academy Postgraduate School at Annapolis, and in 1924 received a Master of Science Degree at Massachusetts Institute of Technology. In World War II he was responsible for the design and development of the Navy's wartime aircraft and airborne equipment, and was awarded the Legion of Merit for his contributions to the advances made by the United States Navy in its naval air program.

Subsequent to World War II, he directed the aviation material activities on the Staff of the Commander of Joint Task Force One, which supervised the atomic bomb tests at Bikini Atoll.

In addition to the Legion of Merit, Admiral Lonnquest holds the Secretary of the Navy's Commendation Ribbon for his work at Bikini. He also holds the Victory Medals of both World Wars and other Campaign and Area Medals.

In 1950, Admiral Lonnquest was elected a Fellow in the Institute of Aeronautical Sciences; an honor accorded only to those who have made notable and valuable contributions to aeronautical science and engineering. In 1948, Admiral Lonnquest received the National Air Council Award "For his vision, intelligence, sound judgment and his dynamic leadership . . . in prosecuting effectively the post-war research and development program of this important agency of our national defense (Navy) and in solving its many complex and difficult problems."

The Secretary-Treasurer for 1952

CAPTAIN J. E. HAMILTON, U.S.N., (Retired), was reelected.

New Council members for 1952-3

REAR ADMIRAL R. E. MCSHANE, U.S.N.
CAPTAIN C. G. GRIMES, U.S.N.
COMMANDER E. C. THOMPSON, JR., U.S.C.G.
CAPTAIN CHARLES BITTINGER, U.S.N.R.
MR. J. J. KENNEDY

Members of the Council whose terms carry over until 31 December 1952 are:

CAPTAIN W. A. DOLAN, JR., U.S.N.
CAPTAIN F. W. WALTON, U.S.N.
CAPTAIN FRED WEISNER, U.S.N.R., and
MR. JOHN C. NIEDERMAIER

The proposed amendment to the by-laws, to provide that appointment by the Council of an Assistant Secretary-Treasurer shall be permissible rather than mandatory, was adopted.

JOINT COUNCIL MEETING

At a joint 1951 and 1952 Council meeting on 3 January 1952, the results of the ballot count were received and results as described above were confirmed. Admiral Lonnquest was inducted into the Presidency at this meeting.

The Council reappointed CAPTAIN ROBERT B. MADDEN, U.S.N., as Assistant Secretary-Treasurer and affirmed with expressions of sincere thanks and confidence the continued employment of MR. A. G. FESSENDEN as Clerk of the Society.

DESIGNATED EDO-ERRATA

It was feared that errors and omissions would be found in the Staff Article which was published in the November 1951 JOURNAL. Two omissions have been discovered to date and it is sincerely regretted that these were not included in the original printing.

REAR ADMIRAL ALBERT T. CHURCH

The article failed to note that Admiral Church served as Secretary-Treasurer of the Society in 1915 and 1916.

LIEUTENANT COMMANDER LOUIS R. FORD

Mr. Ford's name, through error, was omitted from the article. Mr. Ford has been a member of the Society since 28 January 1914. This record of 38 years membership places him in a very select group of those with such long active association with the Society. Mr. Ford was designated for Engineering Duty Only on 10 March 1919 by the Secretary of the Navy Josephus Daniels. He resigned from the Navy with the rank of Lieutenant Commander a few months later.

Mr. Ford holds the distinction, as far as we have been able to determine, of being the only Warrant Officer ever to have been commissioned (1912) and designated EDO (1919) during the period covered by the Staff Article.

SOCIETY BUDGET

The following exchange of letters may help to clarify the Society Budget situation:

ELLIOTT COMPANY
(INCORPORATED)

Jeannette, Pa.
December 7, 1951

Secretary-Treasurer
The American Society of Naval Engineers
605 F St. N. W.
Washington 4, D. C.

Dear Sir:

I have examined with considerable interest, and I might add, amazement, the budget for 1952 as proposed and approved by council. My reaction on comparing the proposed budget with the actual income-expense statement of 1950 operations is that the new budget has been set up unrealistically, merely to prove on paper, that by adopting a budget, a net profit will ensue for 1952's balance sheet!

For example, the following comparison of figures published in the November 1951 issue of the transactions are revealing, and need explanation. Remember that costs have gone up since 1950:

It is estimated that subscriptions, and hence number of journals printed, and advertising will increase approximately 60% over 1950. Yet printing, engraving and manuscript costs are approximately 85% of 1950 costs. Also, the profit and loss will decrease from $5341.50 to $1000.00, although this figure apparently represents delinquent dues which are not easy to collect. An item-by-item comparison of the 1950 vs. 1952 expense and income sheets reveals many more such illogical discrepancies.

No one denies that budget control is desirable and necessary, but certainly it must be realistic to serve a useful function. Especially is this true since this false budget has apparently been used to back the conclusion that increases in dues, subscription rates and advertising costs are not required.

Very truly yours,
J. R. SHIELDS
Member.

December 21, 1951

Mr. J. R. Shields
Elliott Company
Jeannette, Pa.
Dear Mr. Shields:

It was a very pleasant experience to receive your letter dated 7 December, 1951. So few members take enough interest to tell us what they are thinking that it is not easy to know what the members want.

The formal budget is a new experiment and it was rigged very definitely as a control instrument. There was no intent to prove anything. The matter of dues, subscriptions and advertising rates was incidental. As a matter of policy, the Council decided against increasing rates unless we have a real runaway inflation or an inordinate increase in paper or printing or mailing cost.

The intent of the budget was to require the Secretary-Treasurer to hold expenditures down to somewhat less than expected income. As has previously been discussed several times in the Secretary's Notes, the principal means of control is in the cost of the Journal. Since we have been running a deficit for three years and one full year of this has been under the present increased rates, it is apparent that the cost of the Journal must be reduced.

Members may expect that future issues of the Journal will be somewhat smaller than recent issues have been. By a reduction from approximately 1000 to 800 pages per year, with a corresponding reduction in engraving costs, we hope to be able to operate in the black.

If members object seriously to the reduction in size of the Journal, we hope that they will let us know. Continuation of the 1951 volume will, naturally, require an increase in rates, which the Council will undoubtedly approve if the consensus of membership demands. Of course, any increase in dues will have to be submitted to the membership as a change in the by-laws.

As regards your specific criticism of the budget, the following explanation is offered.

(a) The predicted subscriptions for 1952 are exactly the same as at present. The rate is figured at the full rate of $9.00. In 1950 the principal subscription was much less and since it is on a fiscal year basis, half of the 1950 rate was the old one of $6.00. This applied to more than 80 per cent. of the subscriptions. Some advertising contracts also carried into 1950 at the older rate.

(b) The reduction in printing, engraving and manuscript cost has been described above. It is deliberate and forced. It recognizes increased unit cost but a forced reduction in volume.

(c) The profit and loss figure may be weak. However, the 1950 figure was inordinately high because it reflected the post-war clearing of the books.

The Council has required that a quarterly statement be prepared hereafter for its consideration. It may well be that as 1952 develops, the budget will be changed quite a bit. In the meantime, we would like to have the members bear with us in our effort to produce the best Journal with present rates and within the Society's income.

If you have no objection, I would like to publish your letter and this reply in the February issue. It is certainly apparent from your letter that the fiscal situation as presented has not been clear. Perhaps the exchange of letters will tend to clarify the picture.

I am,

Very sincerely,

J. E. HAMILTON

Secretary-Treasurer

THE ANNUAL BANQUET

Plans are going forward for the Annual Banquet of the Society which will be held on Friday, 2 May 1952 at the Hotel Statler in Washington, D. C.

Rear Admiral Homer N. Wallin, Chief of the Bureau of Ships, has agreed to serve as Toastmaster. The principal speaker has not yet been selected.

ERRATA

Acknowledgment: The August 1951 issue of the JOURNAL failed to give complete acknowledgment of the reprinted article "Atomic Energy—Industrial Paradox". This article by Eldon C. Shoup was originally published in the HARVARD BUSINESS SCHOOL BULLETIN.

NATIONAL SCIENCE FOUNDATION

The following press release should be of real interest to readers of the Journal:

The $3,500,000 appropriation for the National Science Foundation in the Supplementary Appropriation Act of 1952, signed by the President this week, will enable the Foundation to start immediately on its two major operating programs: support of basic research in the sciences and training of scientific manpower. Up until now the Foundation has been concerned primarily with planning and with the problems of organization and staffing prior to beginning its operating program.

The reduction in the appropriation from the President's budget request of $14,000,000 for fiscal 1952 has required material scaling down and adjustment of the program originally presented to the Congress. According to Dr. Alan T. Waterman, director of the Foundation, approximately $1,500,000 of the available funds will be allocated for the support of basic research in biology, medicine, mathematics, the physical sciences, and engineering; about $1,350,000 for the training of scientific manpower; and the balance for development of a national policy for the promotion of basic research and education in the sciences, for the wider dissemination of scientific information, and for other services including support of the National Scientific Register, now established in the Office of Education.

Basic scientific research will in most cases be supported by means of research grants. No grants have been made to date, although a considerable number of research proposals have been received from investigators in all parts of the United States. It is expected that many more proposals will be received shortly.

Proposals for research grants will be given preliminary evaluation and review by the Foundation's three research divisions: the Division of Biological Sciences under Dr. John Field, the Division of Mathematical, Physical and Engineering Sciences under Dr. Paul Klopsteg, and the Division of Medical Research under Dr. John Field (acting). Each Division will be assisted in evaluation and review by a Divisional Committee, made up of outstanding scientists in the field, and by expert consultants employed on a part-time basis. Grants will be approved by the Director and the 24-member National Science Board, whose chairman is Dr. James B. Conant, president of Harvard University.

The National Science Foundation Graduate Fellowship program will be directed by the Division of Scientific Personnel and Education under Dr. Harry C. Kelly. Selection of fellows will be made solely on the basis of ability and will be carried on by the National Research Council, which has participated in the selection of fellows for various government and private agencies for many years.

Applications will be considered from students in the natural sciences who have or will have completed their undergraduate work in any accredited college or university. Fellows may attend any accredited non-profit institution offering graduate studies in science which approves their application for admission.

Announcements regarding the National Science Foundation Graduate Fellowships will be distributed within the next week or two. These will describe stipends and allowances in detail. Inquiries and applications should be addressed to the Fellowship Office, National Research Council, Washington 25, D. C.

The Journal

As a departure from past custom, original and reprinted articles will no longer be divided into two separate sections of the Journal. They will be intermixed throughout. Original articles will be distinguished by carrying a by-line and a biography of "the author." Reprinted articles will be headed by "an acknowledgement."

Discussion

We have received a technical criticism of the article "The Wave Drag of Ships" by Max Otto Kramer, which was published on pages 575 to 582 of the August 1951 issue of the Journal. Since time does not permit the preparation of an adequate discussion for this issue, it will be published in May 1952.

JOINT COUNCIL MEETING — 3 JANUARY, 1952

Seated (Left to Right): Capt. F. W. Walton, U.S.N., 1951-52 Council; Mr. A. G. Fessenden, Clerk; Mr. J. J. Kennedy, 1952-53 Council; Rear Admiral Louis Dreller, U.S.N. (Ret.), 1951 President; Rear Admiral T. C. Lonnquest, U.S.N., 1952 President; Mr. John C. Niedermaier, 1951-52 Council; Captain Charles Bittenger, U.S.N.R., 1952-53 Council. *Standing:* Commander E. C. Thompson, U.S.C.G., 1952-53 Council; Commander H. B. Reece, U.S.N., Chairman Ballot Committee; Rear Admiral R. E. McShane, U.S.N., 1952-53 Council; Mr. Francis H. Engel, 1950-51 Council; Captain W. A. Dolan, U.S.N., 1951-52 Council; Captain Frederick C. Wiesner, U.S.N.R., 1951-52 Council; Captain Rawson Bennett, U.S.N., 1951 Council; Captain J. E. Hamilton, U.S.N. (Ret.), Secretary-Treasurer; Captain Robert B. Madden, U.S.N., Assistant Secretary-Treasurer. *Absentees:* Captain Joseph B. Duval, U.S.N., 1951 Council; Captain R. A. Smyth, U.S.C.G., 1950-51 Council; Captain W. L. Tann, U.S.N.R., 1950-51 Council; Captain C. G. Grimes, U.S.N., 1952-53 Council.

SHIP MOTIONS

JOHN C. NIEDERMAIR

This paper was presented in London on 28 June, 1951 before the International Conference of Naval Architects and Marine Engineers. The author who is a member of the Council of the American Society of Naval Engineers was initially sponsored by the Society of Naval Architects and Marine Engineers. By agreement his sponsorship was shared by ASNE and this paper is published in the JOURNAL *as an original paper in the United States. This publication is by agreement with the Society of Naval Architects and Marine Engineers.*

THE AUTHOR

graduated from Webb Institute of Naval Architecture in 1918. Nearly all of his experience since that time has been in connection with the design of U.S. Naval Vessels. Most of his sea experience was obtained on the Atlantic during winter storm conditions. He started his work with the U.S. Navy at the New York Naval Shipyard in 1918. Prior to his transfer to the Bureau of Construction and repair in 1928, he served as technical assistant on the salvage operations of the S-51 and S-4 submarines. In 1929 he was a technical expert with the American Delegation to the 1929 London Conference on Safety of Life at Sea. During World War II he was the Chief Naval Architect in the Bureau of Ships. For his part in the design of Navy ships during World War II he was awarded the Distinguished Civilian Service Award. At present he is the Technical Director of Preliminary Ship Design in the Bureau of Ships.

SUMMARY

Recent knowledge gained concerning the six oscillatory motions of ships at sea is reviewed. These motions are grouped under the following headings: Reduction of speed in waves, Pitch, Surge, Heave, Roll and Directional Stability. New experimental data are presented for the first three subjects which are correlated with recent theoretical findings. Specifically, the following conclusions from theoretical calculations are confirmed:

(*a*) In regular waves the maximum pitching amplitudes do not necessarily occur at synchronism or at wave-lengths equal to the ship length.

(*b*) Absolute motions are less at high than at low Froude numbers when the wave-length is moderate.

(*c*) The maximum amplitudes of motions may be expected at high speeds in very long waves.

A standard sea condition for inclusion in all rough water model testing is proposed together with a formula for estimating average wave heights. The difficulties of full-scale testing are discussed and the need for both new facilities and more theoretical work in the field is emphasized.

1. INTRODUCTION

In the broadest sense, ship motions cover a large portion of the problems with which the naval architect has to work. Hence some breakdown of the subject seems essential in order to be able to select a portion of the problem which can be treated in a brief paper. The following breakdown seems appropriate: (*a*) Desirable steady motions induced by forces which can be exerted from within the ship. Making headway and turning fall into this category.

(*b*) Undesirable oscillatory motions induced by forces external to the ship such as the sea and wind, but over which the ship can exert direct control by means of specially installed devices. Yawing and swaying * always fall into this category and rolling does occasionally.

(*c*) Undesirable motions, oscillatory and otherwise, induced by the sea and wind over which the ship can exert no direct control. Pitching, heaving, surging, and reduction of speed in a seaway, practically always fall in this category.

In the past, the priority of interest by the naval architect in these different categories of ship motions has been roughly the same as the order in which the categories have been listed. At present, however, with increased emphasis on maintenance of speed in a seaway, interest is growing in the motions of category (*c*) particularly, and (*b*) somewhat. This paper will therefore discuss those categories only. Recent papers by Weinblum and St. Denis, (1) and Havelock, (2) have provided excellent summaries of the theoretical treatment of these motions. This paper will deal more with experimental evidence and with observations relating the experimental data to theoretical findings. By this sort of treatment of the subject it is hoped that interest in ship motions by both the naval architect and the physicist will be further stimulated.

* The oscillatory motion nomenclature of Ref. 1 is used here, viz:

Rotation about x–x = roll.		Translation along x–x = surge.	
" " y–y = pitch.		" " y–y = sway.	
" " z–z = yaw.		" " z–z = heave.	

2. REDUCTION OF SPEED IN WAVES

In contrast to the pitching amplitude problem which will be treated later in the paper, the reduction of speed problem has not as yet received adequate theoretical treatment. In an effort to stimulate more work in the theoretical field some experimental data on specific designs will be presented. Some observations can be drawn from this experimental data which might be helpful in accounting for all the sources of increased resistance in rough water.

Much has been written by Kent (3) to (5), on the added resistance in rough water, but it is really very illuminating to study the analogous problem, that of reduction of speed with a constant thrust. Through the facilities and staff of the Newport News Shipbuilding and Drydock Co. Hydraulic Laboratory, the United States Navy obtained some data on that particular problem. This laboratory possesses a small tank, 56 ft. by 8 ft., but what is lacking in size is made up in quality. Towing is accomplished by means of a very carefully engineered falling weight. Equipped with a wavemaker, this facility permits measurements of pitching amplitude, reduction in speed, and the magnitude of surging, all at constant thrust.

The characteristics of the prototypes of the models tested are tabulated in

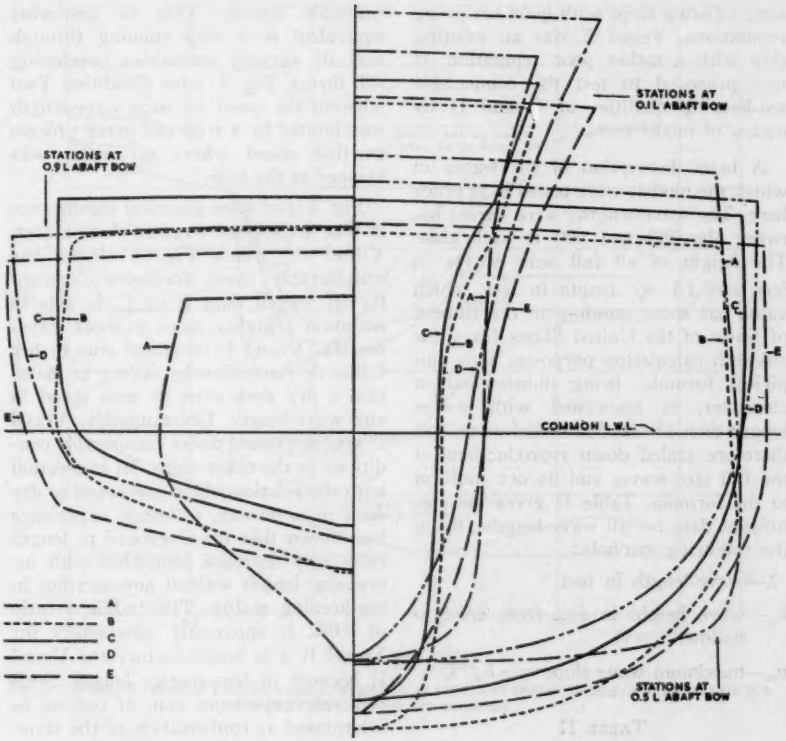

FIG. 1.—COMPARATIVE SECTIONS OF VESSELS "A" THROUGH "E"; ALL REDUCED TO A COMMON LENGTH

Table I and comparative details of form are shown in Fig. 1. It will be noted that the individual designs bear no systematic relationship to each other. Therefore any consistency in the ship motion data of these designs may be construed as generally applicable to hull forms of their type.

The comparative nature of the tests should be emphasized. Vessels A and B

TABLE I

Vessel		A	B	C	D	E
D W L, ft.	L	550	530	520	480	380
Draught ft.	d	15.9	20.0	17.8	14.0	13.5
Max. sect. coefficient	C_m	0.951	0.85	0.818	0.805	0.841
Block coefficient	C_b	0.621	0.513	0.467	0.497	0.535
Water-plane coefficient	C_w	0.707	0.738	0.690	0.739	0.763
Freeboard/length at 0.1 L abaft bow		0.048	0.042	0.044	0.053	0.042
L C B aft ⌀, per cent L		0.5	1.5	0.6	1.9	2.2
L C G of L W L aft ⌀, per cent L		2.1	6.9	6.4	6.2	5.9
Scale ratio: ship/model		120	120	120	100	85

were existing ships with good sea-going reputations. Vessel E was an existing ship with a rather poor reputation. It was proposed to test the comparable sea-keeping qualities of C and D by means of model tests.

A brief description of the waves in which the models were tested is in order here. The wave-lengths were varied between the 300 and 900 ft. full scale. The height of all full scale waves in feet was 1.1 $\sqrt{\text{length in feet}}$, which value has some standing in the Bureau of Ships of the United States Navy for strength calculation purposes. This empirical formula, being dimensional in character, is associated with waves longer than 150 ft. The model waves are therefore scaled down reproductions of the full size waves and do not conform to the formula. Table II gives the significant data for all wave-lengths, using the following symbols:

λ—wave-length in feet.

h_w—wave height in feet from crest to hollow.

v_m—maximum wave slope $= \pi h_w/\lambda$.

TABLE II

λ	h_w	v_m	λ	h_w	v_m
300	19.1	0.199	600	26.9	0.141
400	22.0	0.173	700	29.1	0.131
500	24.6	0.155	800	31.1	0.122
			900	33.0	0.115

It will be noted that the maximum wave slopes decline with increasing length of wave which is in accordance with available information on actual wave heights.

The results of the speed reduction tests are shown in Figs. 2 and 3 in terms of Froude number versus wave-length to ship-length ratio. Fig. 2 is for Condition One, wherein the towing pan was loaded with a weight necessary to pull the model at the stated still water full-scale speeds. This is somewhat equivalent to a ship running through seas of varying intensities developing full thrust. Fig. 3 is for Condition Two wherein the speed for each wave-length was limited by a trial and error process to that speed where no water was shipped at the bow.

Fig. 3 is of some practical significance in that it confirms a rather obvious fact. Vessel A, which Fig. 1 shows has considerably more freeboard forward for its length than B or C, is able to maintain a higher speed in most wave-lengths. Vessel E remained true to her full-scale reputation by failing to maintain a dry deck even at zero speed in any wave-length. Unfortunately, Vessel D was not tested under comparable conditions to the other ships. In connection with the relationship of freeboard to dry deck in a seaway, full-scale experience has shown that the freeboard to length ratio may decrease somewhat with increasing length without any sacrifice in sea-keeping ability. Thus while a ratio of 0.042 is apparently satisfactory for Vessel B it is unsatisfactory for Vessel E because of her shorter length. Such full-scale experience can, of course, be interpreted as confirmation of the statement made earlier, that short waves are steeper than long waves.

Study of Fig. 2 reveals that the maximum speed reduction occurs almost without exception at wave-lengths about equal to the ship length. This condition is not that for maximum pitching amplitudes as will be shown later on. It will also be noted that the synchronous condition, where the period of encounter is equal to the natural period of the ship, as calculated herein, in all cases occurs just prior to the maximum reduction in speed. It may be deduced therefore that either the reduction of speed, like pitching amplitude, is not the most critical at synchronism, or else that the natural pitching periods are slightly longer than have been calculated herein. This latter point will be discussed later. It

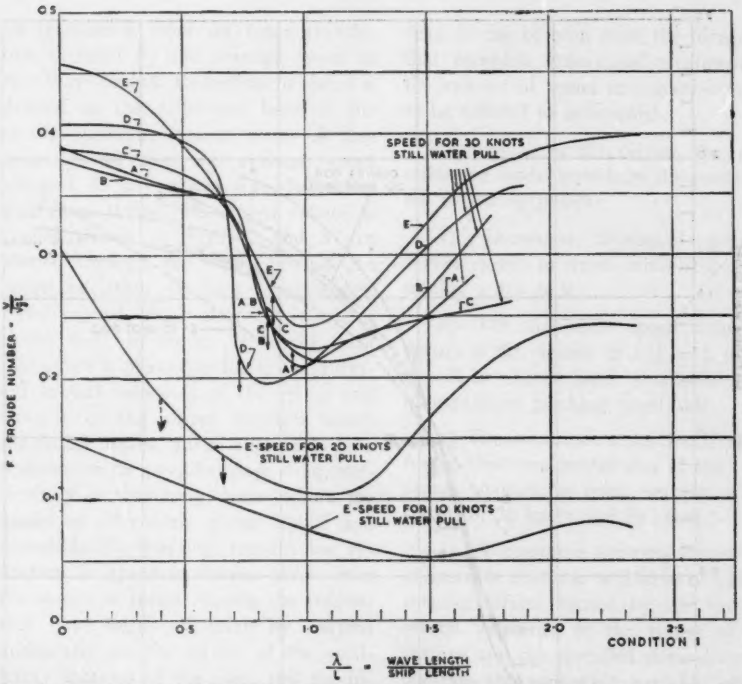
SPEED FOR 30 KNOTS STILL WATER PULL
E-SPEED FOR 20 KNOTS STILL WATER PULL
E-SPEED FOR 10 KNOTS STILL WATER PULL
F = FROUDE NUMBER = V/√gL
λ/L = WAVE LENGTH / SHIP LENGTH
CONDITION 1

FIG. 2.—SPEED, IN TERMS OF FROUDE NUMBER, MEASURED AT CONSTANT THRUST IN SEAS OF VARYING LENGTHS
⟷ INDICATES λ/L FOR SYNCHRONISM

may also be deduced that the exciting forces that cause the maximum reduction of speed occur at far different wave-lengths than the exciting forces which produce the maximum amplitudes of pitch (see Fig. 4). This is contrary to deductions by Kent (5).

Some observations of theoretical interest have been drawn above from Fig. 2. There are also a few practical observations that can be made. The reduction in speed at constant thrust is enormous from smooth water conditions to seas of wave-lengths on the order of a ship length. In terms of knots the reductions in speed are shown in Table III.

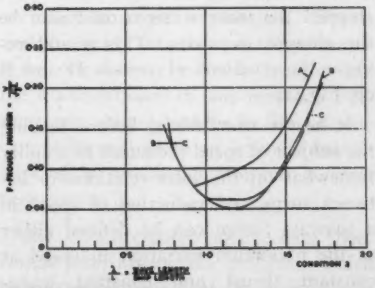

FIG. 3.—SPEED, IN TERMS OF FROUDE NUMBER, AT WHICH WATER COMES OVER THE BOW IN SEAS OF VARYING LENGTH

TABLE III

Vessel	A	B	C	D	E	E	E
Still water speed	30	30	30	30	30	20	10
Speed in worst sea condition	17	17.5	16	14.5	16	7	3
Reduction in speed	13	12.5	14	15.5	14	13	7

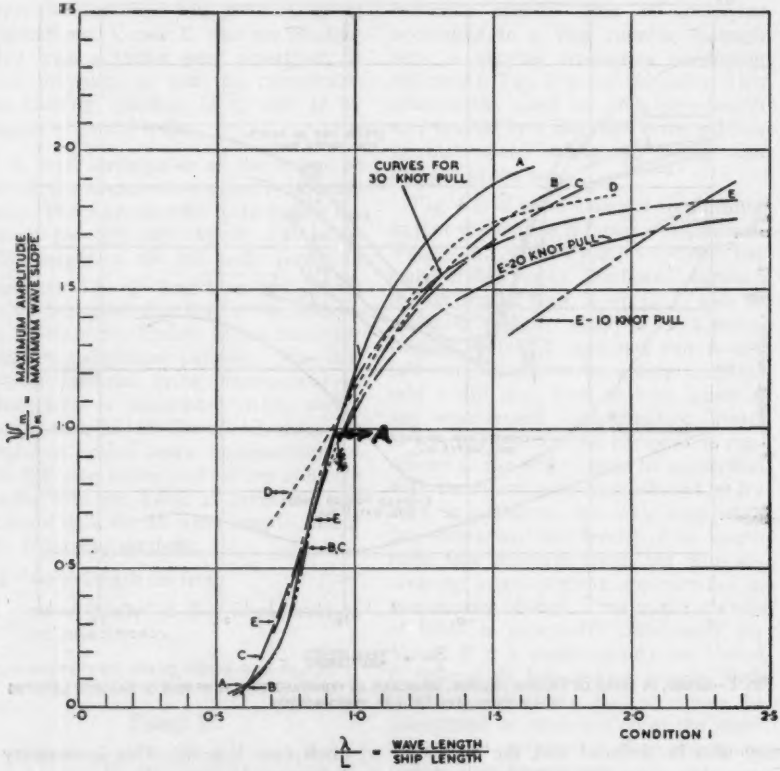

FIG. 4.—MAXIMUM PITCHING AMPLITUDE/MAXIMUM WAVE SLOPE FOR INDIVIDUAL MODELS IN SEAS OF VARYING LENGTH. ⟷ INDICATES λ/L FOR SYNCHRONISM

One interesting item is that the reduction in speed from 20 knots smooth water speed is of the same actual magnitude as the reduction from 30 knots. Another observation is that vessel D, which according to Fig. 1 has considerably more freeboard than E, suffered greater speed reduction at constant thrust. Visual observation of the tests showed, however, that D was vastly drier than E, as would be expected. The models discussed herein had no superstructure built on. It may be that with a superstructure the model that shipped the most water would also be the slowest in waves. This would reverse the positions of vessels D and E on Fig. 2.

It seems worthwhile, before leaving the subject of speed reduction to amplify somewhat on the inter-relationship between surge and reduction of speed in a seaway. Surge can be defined either as the maximum variation in speed at constant thrust and constant wavelength, or as the variation in fore and

aft translation from the mean translation dictated by the average speed in that wave-length. Reduction in speed is defined as the difference between the steady speed in smooth water at any given thrust and the average speed attained in any given wave-length at that same thrust. The speed reduction curves shown in Figs. 2 and 3 are plotted through the mid-points of the surge variation at each wave-length. From visual observation of constant thrust tests in waves, it would seem that surge is primarily due to the different orbital velocities of the crests and troughs of the waves through which the model passes. Surge might also be a function of the so-called slope drag, particularly in very long waves where the model is alternately going uphill and downhill. The physical reasons for reduction in speed in waves differ from the causes of surge. Among the reasons that have been put forth by various authorities are the effects of the oscillatory motions of the ship, and the interference effects between the wave pattern which the ship seeks to generate and the wave pattern of the surrounding seas. It can be seen from the foregoing that complete theoretical treatment of the subject of speed in waves is going to be difficult to accomplish.

In summarizing this section, the speed reduction model tests have demonstrated the following points:

(*a*) Increased freeboard permits higher speeds in waves while still maintaining a dry deck.

(*b*) The maximum speed reduction occurs in the vicinity of $\lambda/L = 1$, which it will be shown later is not the point of maximum pitching amplitude.

(*c*) The maximum reduction in speed for at least one model was of the same actual magnitude from smooth water speeds of 30 knots and 20 knots.

(*d*) A complete theoretical analysis of speed in a seaway will have to include surging effects caused by the varying orbital velocities of the waves at the surface and the so-called slope drag. In addition the somewhat variable effects of oscillatory motions of ship and wave interference will also have to be considered.

3. PITCH

The maximum amplitude of pitch has not been a very vital figure for the naval architect to know in the past. Nevertheless, study of the pitching motion gives one a very valuable insight into the whole problem of ship motions. Furthermore, with advances in the art, one cannot foretell when an intimate knowledge of the pitching motion will be most necessary.

Ref. (1) presented an excellent theoretical analysis of the problem, which led to conclusions that, in many cases, were not in accordance with several commonly accepted rules. In an effort to shed further light on these conclusions, some of the evidence obtained from the experiments at Newport News was applied to the pitching amplitude problem. In most cases the pertinent theories of Ref. (1) were found to be in substantial agreement with these new experimental data.

Perhaps the most interesting conclusion of both Refs. (1) and (2) is that maximum pitching amplitudes do not necessarily occur at synchronism. This paradox is explained on the basis that

the maximum amplitudes are quite as much a function of the magnitude of the exciting forces of the seas as they are of the relationship between the natural period of the ships and the period of encounter with the waves. The exciting forces in general grow with increasing wave-lengths and increasing speed. At the point of synchronism when the period of encounter with the seas is the same as the natural period of the ship, the exciting forces have by no means reached a maximum. Therefore, in many cases, the pitching amplitudes are not a maximum until wave-length well beyond those corresponding to synchronism, are reached. That this conclusion is substantially in agreement with experimental data will be shown a little later.

The natural pitching period of the vessel has been mentioned frequently. Yet this is a very controversial figure, as borne out by Ref. (2). The figures used here, in the absence of any full-scale data, have been calculated on the basis of the figures supplied in Ref. (1), particularly in regard to the inertia of the entrained water or the added inertia. Fig. 10 of Ref. (1) gives the coefficients of accession to inertia for the pitching motion which, when applied to the mass moment of inertia of the volume of a body, give the inertia of the entrained water. The data are for deeply submerged bodies, but lacking data for bodies at the free surface, it is used herein. No great accuracy can be claimed for these inertia coefficients, but it has nevertheless proved interesting to calculate the moment of inertia of the ship itself plus the inertia of the entrained water with some care. Table I and Table IV list the data used in the calculations. It will be noted that the inertia of the weights has been calculated in addition to the inertia of the volume. The formula as given in Ref. (1) for the natural period of pitch, $T\psi$, is as follows:

$$T\psi = 2\pi \sqrt{\frac{I_y + I_{yy}}{\rho g J_y}}$$

where I_y = mass moment of inertia of the weights.

I_{yy} = added mass moment of inertia.

J_y = longl. moment of inertia of the water plane.

which was converted to the following for our use:

$$T\psi = 2\pi/K_3 \sqrt{\frac{d \times C_b}{g\, C\, w} (K_1^2 + k_{yy} K_2^2)}$$

with symbols as defined in Table I and IV.

TABLE IV

Item	Symbol	A	B	C	D	E
Gyradius of weight/L	K_1	0.227	0.220	0.227	0.225	0.233
Gyradius of volume/L	K_2	0.215	0.211	0.202	0.203	0.216
Gyradius of water-plane/L	K_3	0.227	0.234	0.229	0.240	0.241
Coefficient of accession to inertia*	k_{yy}	1.48	1.12	1.30	1.48	1.31
Natural period in secs.	$T\psi$	6.31	5.51	5.34	4.66	4.77

* Taken from Ref. (1), Fig. 10.

Because of the fact that K_2 is less than K_1 for the vessels considered, the net addition to inertia is not quite as great as if k_{yy} were applied to the inertia of the weight as is sometimes the case. As a result of treating the added inertia in the way indicated by the formula and Table IV, the net increase in the natural period over that calculated by neglecting the added inertia, ranges from 42 percent for vessel B to 53 percent for vessel A. Fig. 2 shows an example of where synchronism would occur had the shorter period, obtained by neglecting the added inertia, been used herein. The dotted arrow for vessel E at 20 knots still water pull shows that location of synchronism. It is obvious from Fig. 2 that the dotted arrow is even further from the point of minimum speed than the full arrow. It may therefore be true, as mentioned in the earlier discussion of Fig. 2, that the true added inertia coefficients are even greater than those of Ref. (1) used herein. Such a possibility does exist, since the added inertia coefficients may very well be a function of factors neglected herein, such as, free surface effects and speed of advance. In any case, it seems safe to assume that the actual natural period of pitch of the ship is much closer to those of Table IV than it is to the period calculated by neglecting the added inertia of the entrained water.

The pitching amplitudes of Vessels A through E are shown in Fig. 4 as ratios of maximum amplitude to maximum wave slope plotted against wave-length to ship length ratio. The amplitudes are for Condition One which, as stated previously, corresponds to constant thrust for a given still water speed. Thus the speed to be associated with each wave-length is not constant but varies as shown on Fig. 2. Because of the variation in speed, these amplitude curves cannot be compared directly to those of other authors which generally have been associated with a constant speed. Several observations can be made concerning these curves:

(*a*) For the same still water speed pull the curves for all ships fall surprisingly close together.

(*b*) The difference between the curves for Vessel E at various still water pulls are greater than the difference between individual ships at the same still water pull.

(*c*) The observations of (*a*) and (*b*) strongly attest to the theme that the maximum pitching amplitudes are much more a function of the exciting forces of the sea than they are of the individuality of the ship. The exciting forces are, of course, a function of the size of the waves and the speed of the ship.

(*d*) The amplitudes approach a maximum at high wave-lengths. The conditions of either synchronism or $\lambda/L = 1$, appear to be of little significance as far as maximum amplitudes are concerned. This observation may only be applicable to the particular condition of regular waves.

Observation (*a*) leads to the possibility that the amplitude data for all of the different models can be treated as if it were obtained from a single model. Using that assumption, the data were segregated around constant speeds irrespective of the model to which the data pertained. The results of this approach are shown on Fig. 5 as contours of constant Froude number on coordinates identical to those of Fig. 4. Comparison of Figs. 4 and 5 demonstrates once again that the effect of speed on the amplitude of pitch is more pronounced than the effect of the different hull forms of vessels A through E. It will be noted on Fig. 5, as on Fig. 4, that the arrows indicating synchronism again occur at wave-lengths far from the maximum pitching amplitudes. Fig. 5 also indicates that in wave-lengths of λ/L less than 1.3, the absolute pitching amplitudes are somewhat smaller at the

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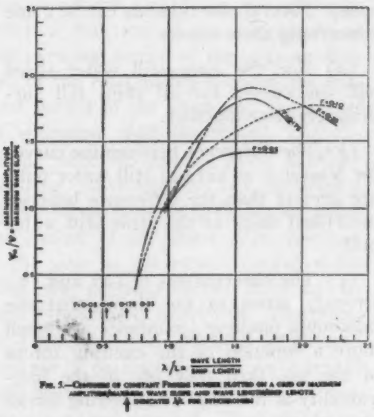

FIG. 5.—CURVES OF CONSTANT FROUDE NUMBER PLOTTED ON A GRID OF MAXIMUM AMPLITUDE/MAXIMUM WAVE SLOPE AND WAVE LENGTH/SHIP LENGTH RATIOS. ARROWS INDICATE $\lambda/L = 1$ FOR SYNCHRONISM

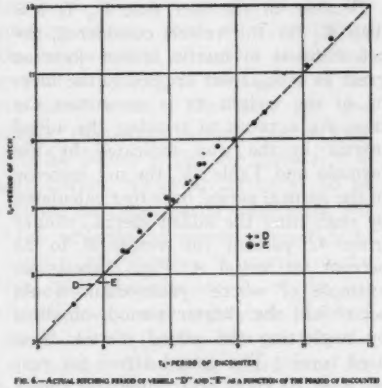

FIG. 6.—ACTUAL PITCHING PERIOD OF VESSELS "D" AND "E" AS A FUNCTION OF THE PERIOD OF ENCOUNTER

highest speed shown than at lower speeds.

Actual pitching periods were measured on vessels D and E for all of the tests of Condition One. These actual periods are plotted on Fig. 6 against T_e, the period of encounter. It is seen at a glance that the two periods are practically coincident for the whole range of waves and speeds in which the tests were run. The arrows on Fig. 6 indicate the natural pitching period of the ships. The fact that there is no gathering of actual pitching periods in the vicinity of the natural pitching periods indicates that the pitching for these tests was forced. This points up the dissimilarity between model tests in regular waves and an actual ship at sea. According to Ref. (6), a ship at sea pitches in her own natural period a large proportion of the time because of the irregularity of the seas. It seems safe to assume, however, that on the few occasions that a ship does meet large regular waves it, too, would pitch in the period of the large waves.

There are several conclusions about the pitching motion which are summarized as follows:

(*a*) In regular waves the maximum pitching amplitudes do not necessarily occur at synchronism or at wave-lengths equal to the ship length.

(*b*) Indications are that the calculations for natural pitching period should include an appreciable allowance for the added inertia of the entrained water.

(*c*) Maximum pitching amplitudes in regular, steep waves are much more a function of the speed of the ship and the length of wave than of the individuality of the ship. This conclusion, of course, can be strictly applied only to the range of hull forms shown in Fig. 1.

(*d*) There are indications from the model tests that in wave-lengths of λ/L less than 1.3, the absolute pitching amplitudes are somewhat smaller at the highest speed shown than at lower speeds.

(*e*) The actual pitching period of models in regular waves is practically identical with the period of encounter.

The above conclusions, based on model experiments, agree with most of the pertinent deductions and conclusions listed on pages 35 and 42 of Ref. (1).

4. SURGE

The term surge as used herein is defined as the maximum variation in speed at constant thrust and constant wavelength. Ref. (1) states that the maximum amplitude of surge is proportional to the square of the period of encounter. Unfortunately the amplitude of surge as defined there is amplitude of translation rather than amplitude of speed. With the information at hand it is impossible to relate the two definitions. Table V tabulates the surge data as obtained for Condition One for whatever use can be made of it.

TABLE V

T_e = period of encounter in seconds.

ΔV_m = maximum variation in speed in terms of Froude number.

VESSEL									
A		B		C		D		E	
T_e	ΔV_m	T_e	ΔV_m	T_e	ΔV_m	T_e	ΔV_m	T_e	ΔV_m
3.50	0.012	3.54	0.013	3.50	0.005	3.49	0.032	4.10	0.027
4.42	0.006	4.59	0.014	4.75	0.012	5.76	0.018	5.56	0.066
6.13	0.017	6.24	—	6.44	0.026	6.44	0.020	6.10	0.027
7.16	0.013	7.11	0.032	7.07	0.026	6.87	0.026	6.41	0.016
7.87	0.013	7.68	0.026	7.67	0,043	7.22	0.017	6.87	0,015
8.37	0.007	8.08	0.020	8.33	0.009	7.60	0.032	7.50	0,031
8.65	0.005	8.29	—	8.85	0.009	8.04	0.032	8.03	0.023

5. HEAVE

The data from the tests described here have not been analyzed to the extent of obtaining heave data. Ref. (1) has, however, brought out the similarity of the motions of pitch and heave. It may therefore be surmised that the conclusions arrived at for the pitching motion apply with equal verity to heaving.

6. ROLL

Of all the undesirable motions of a ship, rolling is perhaps the most undesirable. For that reason, if for no other, the study of rolling has progressed further in years past than study of the other motions. The intimate relation of the rolling characteristics of ships to their ability to remain upright after damage and flooding has also contributed to the emphasis on roll. The author in Ref. (7) proposed a criterion for GM, metacentric height, the measure of a ship's static restoring force, which was considered to be a satisfactory compromise between the conflicting demands of adequate safety at sea and easy rolling characteristics. This criterion, that the GM should be about 6 percent of the beam for merchant vessels, has been used in the design of a number of large

passenger ships whose subsequent operating experience has been reported as quite satisfactory. It is heartening to hear these reports for there were many who phophesied that GMs of such magnitude would result in highly uncomfortable rolling motions.

The severity of rolling on ships has resulted in the actual installation of various devices to reduce rolling. Among these was the famous gyro stabilizer, whose era already seems past. Currently there is great interest in the Denny-Brown activated fin stabilizer, which, by utilizing a very small portion of the propulsion power of a ship, produces large stabilizing forces. These stabilizers combined with modern intelligence instruments, which can predict the instant of encounter with a wave, offer a great promise of being highly successful roll inhibitors.

7. DIRECTIONAL STABILITY

The remaining possible motions of a ship at sea, yawing and swaying can best be discussed by combining them into one consideration, directional stability. This is so because the two motions are so interdependent, and the hydrodynamic forces which affect both are so related, that any effort to separate consideration of the two motions is extremely arbitrary. It is, moreover, most illuminating to consider these motions in terms of a ship's ability or inability to shake off the effect of disturbing forces which cause the motions. In the first case the ship is called directionally stable and ultimately without any corrective action restores itself to a new, straight course. In the second case the ship is directionally unstable and never restores itself to a straight course without some corrective action. The hydrodynamic forces and moments which combine to determine whether a ship is stable or unstable are functions of:

(*a*) The actual magnitude of the angle of attack produced by the yawing and swaying motion (lift).

(*b*) The rate of change of angle of attack with time (the forces which are functions of this derivative are generally termed damping forces).

(*c*) Higher order derivatives of the attack angle.

Ref. (8) derived a mathematical directional stability criterion which is a function of the above hydrodynamic forces and moments. Consideration of this criterion, since the publication of Ref. (8), has gradually influenced the design of certain types of vessels where course keeping is of primary importance.

Another development in the field of directional stability, in recent years, has been the demonstration by means of actual trajectory calculations, that adequate directional stability and prompt response to rudder layover are not incompatible quantities. They are actually somewhat complementary to each other. This does not imply that the ultimate turning circle of a very directionally stable ship will be as small as the circle for a less stable ship. It does mean that the more stable ship may accomplish a small change of heading faster than the less stable ship, all other things being equal. Establishment of this as a fact will lift the burden of one more compromise from the shoulders of the naval architect.

The entire subject of directional stability is receiving considerable attention today. The brief review given here is meant principally to highlight a few of the more important considerations and developments.

8. CONCLUSIONS

An attempt has been made in this paper to relate the results of some meager experiments to recently published theoretical results. In general the model experiments and the theory agree. It is also probably safe to say that a ship suffering from the same regular seas that the theory and model experiments presuppose, would behave similarly. It cannot be said that the model tests fulfilled in every respect the original purpose of running them. For in spite of some rather wide whole differences between individual models, all of the models behaved substantially alike. However, verification of this very fact, confirming as it does the theoretical calculations, rendered these model tests well worth while.

The conditions of the model tests were extreme and it is not too surprising that the enormous exciting forces of the waves completely wiped out differences in hull form. These tests do not, therefore, preclude the necessity for studies of hull form variations in the somewhat less severe waves which will be discussed next. But if the lessons of the tests described in this paper are applied, future tests will take the form of systematic hull variations varying one factor at a time.

As intimated in the previous paragraph, indications are that the wave heights used in connection with the experiments discussed in this paper are excessive and unrealistic when compared with actual experience at sea. Furthermore, as mentioned earlier, actual sea conditions seldom include regular series of waves. Most of the time the seas are quite irregular and the excessively high waves occur only during a small percentage of the time. For strength purposes the wave height of $1.1\sqrt{\lambda}$ is satisfactory, as such heights are no doubt encountered from time to time, but for ship motion studies, more nearly average conditions of the sea should be the criteria. For this purpose it is believed that wave heights equal to $0.55\sqrt{\lambda}$ should prove more satisfactory when attempting to predict ship performance from model tests.

It is desirable to ensure that there will always be at least one common sea condition for all rough water model testing. A sea condition having a wavelength of 600 ft. and a wave height of 27 ft. is suggested as a reasonable standard. In addition it will generally be desirable to test in a sufficient number of other wave-lengths, considered in relation to the ship length, to cover the range of maximum speed reductions. Such waves could very well be limited to heights equal to $0.55\sqrt{\lambda}$ for the full-scale wave.

The validity of the arbitrary wavelengths and heights suggested in the last paragraph can be partially checked by statistical studies of ocean waves. Such studies could be limited to relatively well traveled ocean routes but should be taken continuously over a period of a few years. Some studies in that direction for the Pacific Ocean have already been accomplished in Ref. (9), but such data need to be broadened to include specifically the factors mentioned herein. Furthermore, similar studies for other ocean routes need to be undertaken.

Measurement of full-scale ship motions at sea plus simultaneous recording of all essential information needed to relate those motions, is a problem fraught with difficulties. Nevertheless some measurements of such nature are probably needed for two reasons:

(*a*) Verification of the validity of the assumed wave conditions in which models are tested.

(*b*) Verification of the similarity of ship and model motions, given geometrically similar wave conditions.

Information of value for (*a*), particularly in regard to reduction of speed in a seaway, can be obtained from sources such as Ref. (10). More data can perhaps come from statistical analysis of the logs of scheduled ships traveling the same ocean route over a long period of time. The average speed figures for any given time of year can then be compared with average wave conditions for that route, obtained as indicated in the last paragraph. It would also be desirable to instrument ships traveling well studied routes in order to obtain rolling and pitching data. There can be little doubt that such studies would yield usable full-scale data for both the designer and the researcher.

Information to satisfy (*b*) above is even more difficult to obtain. It would involve instrumenting a ship fully, both to record ship motions and to record simultaneously the surface of the sea. The instrumentation to make the latter recordings has not as yet been developed and would undoubtedly require the services of trained specialists. Such an instrumented ship with specialists on board could only fulfill its mission on the few rare occasions when it met regular seas. It is believed that there is not as yet sufficient evidence of serious scale effect in the problem of ship motions to warrant such full-scale testing.

The differences between the wave conditions in which ships operate, and the wave conditions in which models are tested has been emphasized. This leads to the argument that the exciting forces of cross seas and irregular seas should be studied in model size in addition to the regular ahead seas currently studied. Simulation of such seas will of course require new extensive facilities. Such facilities would also permit complete six-degree freedom for model experiments. Thus studies of coupled motions, directional stability, and turning could all be further pursued in such a facility.

Of even more importance to the problem than more model tests, full-scale testing, or new facilities, is the continued growth of theoretical developments in the field. We are fortunate in having several eminent scientists interested in ship motions. Their continued efforts will provide the theory so essential for proper guidance of future experimental programs.

9. ACKNOWLEDGMENT

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Discussion

Some of the discussions presented at the meeting in London are given in the following as they add additional opinions and data of value on the subject.

1. MR. R. W. L. GAWN, R.C.N.C. (Member of Council, I.N.A.):

The author remarks that any lack of size of the small tank in which the experiments described in the paper were carried out is made up in quality. Speaking as one who has had the privilege of seeing the Newport News tank his view is fully endorsed. Further, he has covered a wide field of a most important subject with commendable brevity in a paper of undoubted quality.

There is much to be said for investigating ship oscillations by small models. The author's models were nearly 5 ft. long. Some experiments of this character at Admiralty Experiment Works, Haslar, were described in a paper by R. E. Froude as long ago as 1905. The early models then tested were about the same length as those of the author, but the later models were about 7 ft. long, experience having shown that a larger model could be safely ventured on. One

of the considerations which determines the size of models for tests head on to waves is the number of waves which the model meets during the experiment. Ship oscillations generally build up and die down, and in order that the results may be reliable it is necessary that a few cycles of the motion be completed during an experiment. The length of the tank mentioned in the paper is only eleven times the length of the model, and the number of waves which the model meets is limited accordingly. This particularly applies to the longer waves which are of greatest interest. Perhaps the author would remark on this and give his view as to any reaction on the results.

The early experiments at Haslar to which I have referred were made with the model advancing at uniform speed. The author's tests were with the model subjected to uniform towing pull. The two methods are appropriate to the equipment in the respective tanks, but there is something to be said for a constant pull in the sense of perhaps a closer approach to ship conditions. One consequence is, however, that the speed of advance varies during the run, and this departure from steady conditions is a disadvantage from the strictly scientific point of view of comparing with theory.

The author shows in Fig. 2 that the maximum loss of speed occurs in waves of length just less than the ship length. Although the motion is then fairly large it is by no means a peak. Froude's results showed similar phenomena as regards the maximum resistance and the peak resistance covered a comparatively restricted range of wave length.

The curves in Fig. 6 show that the peak value of the amplitude of pitch is nearly double the maximum wave slope. The 1905 results referred to showed that the pitch amplitude rarely exceeds the wave slope, and this is confirmed by later tests, some results of which were given by the speaker in a discussion on the paper referred to in the author's reference 1. Theoretically the amplitude in very long waves will equal the wave amplitude. It is possible that the author has recorded the out to out amplitude of pitch rather than the amplitude each way. Perhaps he would remark on this, because if his curves do relate to the latter amplitude then the angles of pitch recorded are alarming.

The coincidence of pitching period with period of encounter revealed by Fig. 6 is in keeping with the 1905 results, and indeed later investigations in waves. It is very interesting to have this striking confirmation between the results of tests with a constant pull and those with a uniform speed of advance.

Another interesting confirmation is the conclusion on page 9 that all the models behaved substantially alike despite rather wide differences in hull form. It does appear that moderate variations in loading as effecting radius of gyration have a greater influence on the motions than fairly wide variations of hull form. It would be of interest to know if the author's experience bears this out.

The author's definition of natural period on page 5 includes the virtual mass. He assumes that the inertia coefficients can be calculated from the data for deeply submerged bodies, but wisely states that no great claim for accuracy can be made for this assumption. I agree with him that the procedure is controversial. The allowance for virtual mass as determined by him may be seriously in error for a floating body, and no consideration has been given to the influence of damping or speed on the period. Perhaps the author would kindly state if any results are available for the still water pitching period of the models to determine the influence of the various factors. Some results given by the speaker in the discussion on Professor Havelock's paper, mentioned in refer-

ence 2 by the author, indicate that the difference may be large. The pitching period of the ship as distinct from the period of force oscillation is also subject to variation with the size of wave which the ship meets. The only period that can be accurately calculated is that estimated from the fore and aft radius of gyration and longitudinal metacentric height. This quantity is frequently referred to as the natural period. There is much to be said for this as a basic period since a definition which includes a rather arbitrary allowance for inertia coefficients and omits the influence of other factors is a compromise lacking in physical significance.

2. PROFESSOR A. M. ROBB (Vice-President, I.N.A.):

In this paper there is an element of contradiction which deserves attention. In Section 8, at the beginning of the author's "Conclusions," he states that in general the model experiments and the theory agree. In the right-hand column above Fig. 6 he states that in regular waves the maximum pitching amplitudes do not necessarily occur at synchronism. Now, insofar as the importance of synchronism is a central conclusion in any theory of oscillation, we have in these two statements a marked disagreement. On the other hand, in Fig. 6 you will see that the actual pitching period is closely the same as the period of encounter. Theory suggests the conclusion that a major constituent of any oscillation is the forced oscillation in the period of the disturbing force or moment, and therefore in Fig. 6 we have some justification of the theory.

But the disagreement first noted leads to the question whether the free period was ever correctly determined. Mr. Gawn has made some reference to the expression for the pitching period, and has questioned the amount allowed for inertia of entrained water. I had hoped to do the same thing, and I would refer Mr. Niedermair to the only evidence which I can trace on the subject. It is true that it is a case of rolling, but there is an analogy. I refer him to the experiments carried out on the *Revenge*, recorded in the TRANSACTIONS OF THE I.N.A. for 1895. He will find it very difficult to justify any material addition for the inertia in still water if he studies the records of those experiments.

The point here is that with these very small models the period must have been or ought to have been in the region of half a second. This figure is based on Table IV, with an allowance for scale. There is no indication of how the free period was determined, and I should like some information on that.

There is one general point. I suggest the presentation of the results has been considerably affected by the choice of an improper parameter. I refer to Fig. 4, which I suggest is the vital figure of the paper. There the maximum pitching amplitude is related to the maximum wave slope. In fact there is no such direct relation. In the case of rolling we do take the maximum wave slope as a parameter; but we have started with the assumption that the breadth of the ship is small in relation to the length of the wave. Here we have a length of ship which ranges from a major fraction of the wavelength to a multiple of the wavelength, and in that condition maximum wave slope ceases to have any direct significance.

The point is that in the case of rolling the maximum wave slope is also the maximum statical angle of equilibrium; and that is the basic quantity. In the

case of pitching we should also take the maximum statical angle of equilibrium. In that case, instead of the angle decreasing as it does with increase of length of wave in Table II, we shall get an angle which increases with length of wave. Using that more rational parameter, we shall get a totally different result for Fig. 4 and a more rational plotting.

3. DR. F. H. TODD (Member of Council, I.N.A.):

The problem of ship motions is intimately bound up with that of the seaworthiness of ships. The progress of knowledge on this subject has languished in the past because there has never been an adequate theory of the basic elements with which to interpret the results obtained either from models or ships, or to guide us in planning experimental research. This subject of seaworthiness has recently been given quite high priority in the work of the David Taylor Model Basin at Washington, and the recent paper to which Mr. Niedermair refers, by Weinblum and St. Denis, members of the staff of the Basin, will, I think, become more or less a classic guide to the subject of seaworthiness. Mr. Niedermair has given us today some results of model experiments in waves which show general agreement with the theoretical work done by Weinblum and St. Denis.

One of the points the author brings out is the effect of freeboard, not in the accepted sense of that amidships, but at the fore end of the ship. Some model experiments which I carried out some years ago in this country with forms of large trawlers brought out very clearly the immense value of the height of deck at the forward end in ensuring good sea performance. Those experiments also showed that with the relatively high speed/length ratio of these trawlers, one could adopt quite fine angles of entrance and in that way ensure not only good sea-speed qualities, but also dryness of decks. This form, having a fine load water line forward, associated with the correct flare, was as a matter of fact a very successful design. It gave rise to a great deal of opposition in the first instance by ships' captains and owners, who were quite certain that the fining of the fore end to the extent proposed would make a bad sea boat which would not rise to the waves. But they were proved to be wrong; these ships were good from the point of view of seakindliness, and in fact they became very popular.

Model experiments on rough water performance have been in progress for a number of years, one of the chief experimenters in this field being Mr. Kent, at Teddington. All such experiments, however, have been very restricted; the only tests possible have been those in head-on and following seas, and those are conditions we seldom meet in practice. When crossing the Atlantic the sea always seems to me to consist of lumps of water being thrown about, rather than of a particular wave system, and I strongly endorse the author's proposal for new facilities in which models can be run under all conditions of waves.

In the left-hand column of the text above Table IV, the author states: "From visual observation of constant thrust tests in waves, it would seem that surge is primarily due to the different orbital velocities of the crests and troughs. . . ." That is very obvious when we run models in long waves. In such cases the waves are very long compared with the length of the model, so that the orbital velocities are high;

as the model is run down the basin it is very clear how she slows up through the crest and accelerates through the trough.

The author also states that: "Among the reasons that have been put forth by various authorities are the effects of the oscillatory motions of the ship, and the interference effects between the wave pattern which the ship seeks to generate and the wave pattern of the surrounding seas." There again, when these tests are observed in the basin, we find that with relatively fast ships the wave interference is not at all obvious ahead of the models, but with slow ships we can actually see the reflection of the waves from the bows of the model moving away in the forward direction and completely altering the sea pattern the model is meeting.

The models used in the experiments described in the paper are only 4 to 5 ft. long, so that they are quite small. The forces with which we are dealing in these experiments are mostly inertia forces and the effects of pressure distribution on the hull, which are not subject to the same degree of scale effect as the viscous forces we deal with in resistance. We believe that, with proper care and precautions, we can get extremely valuable comparative results from the use of small models. The application of the results to predict actual ship performance is another question, and I believe we need some fundamental work on ship-model correlation in waves, just as we do in resistance work in smooth water. To this end we have recently sent three parties from the Taylor Model Basin staff to sea in three ships on the North Atlantic. Two of those parties have been making voyages across the Atlantic in winter, and one party has spent a month at sea on a coast-guard ship. The ships were not completely instrumented, but we obtained records of powers and speeds, pitching and heaving motions and accelerations. I believe the general observations made will be extremely useful to us in planning further work along these lines.

The subject of seaworthiness is very much alive in the United States at this time, and the American Towing Tank Conference has a very active committee working on it. It has recently been agreed in that committee that we should do some systematic resistance experiments on a number of merchant ship forms varying from 0.60 to 0.80 block coefficient. The Hydromechanics Sub-Committee of the Society of Naval Architects have endorsed the suggestion, and we are in process of setting up a co-operative scheme whereby the different model basins will each take one of the parent forms and make changes in such things as freeboard, flare above water lines, type of sections forward, and so on. At the same time, similar experiments are being undertaken at the Taylor Model Basin with some simplified mathematical forms. Dr. Weinblum, who has associated with Mr. St. Denis in the paper to which Mr. Niedermair has referred, is anxious to determine inertia and damping coefficients for different types of hull. It is much easier for him to work on the results if the hulls used are of mathematical form. As a start we have therefore devised two mathematical forms which have wall sides, and we are putting work in hand to determine some of the coefficients which Mr. Gawn has mentioned, such as the virtual mass and the damping coefficients.

I agree whole-heartedly with Mr. Niedermair in the belief that it is vital to further progress, both to enable us to interpret results and to plan future work along rational and economic lines, that we must develop the theoretical work as far as we can. We can save ourselves much time and money in experimental work if we have a firm theoretical basis for our proposals.

4. PROFESSOR A. J. SIMS, R.C.N.C. (Member of Council, I.N.A.):

On several occasions during this Conference we have been able to take stock of our progress in major directions in naval architecture and marine engineering. It is appropriate that at such an international gathering we should include the subject of "Ship Motions," for it is of such greatly increased importance at this time. We are rightly reminded by the author that the study of rolling has progressed further in years past than has the study of the other ship oscillatory motions. There are good reasons to account for this. The time has come, however, when we must be able to predict more accurately the characteristics of pitching, heaving and loss of speed in a seaway, and this paper is an important step forward to that goal. I propose to confine my remarks to the subject of pitching.

Reference has already been made to the controversial nature of the pitching period. There has been some vagueness in the past in the figures given and the formulae employed. The author has left us in no doubt as to the period he means, and I wish the same could be said about all authors of papers and books on the subject. These are days of standardization of nomenclature, and we should clearly distinguish between three pitching periods:

(a) The "Basic Period" which can be determined by the formulae on page 5 of the author's paper but omitting I_{yy}. This ignores virtual mass effects: it also ignores damping. It is a period which can readily be determined in the design office knowing the longitudinal distribution of weight and the hydrostatics of the ship's form. It is, however, of limited use.

(b) The "Still Water at Rest Period" —at rest meaning that the ship has no ahead motion. This is given by the formulae of the author on page 5 except that it is desirable that the expression given should be divided by $\sqrt{1 - K^2}$. K is the damping factor in the usual damped harmonic equation

$$\frac{d^2\theta}{dt^2} + 2K\,p\,\frac{d\theta}{dt} + p^2\,\theta = 0.$$

Incidentally the K used here is half the value of that used by Weinblum and St. Denis in the author's reference (1). This period is a unique period for any ship, although it is more difficult to determine than the previous one. It is up to half as great again as the "Basic Period."

(c) The "Underway Period" which is given by the same formulae as (b). Changes, however, will take place in three components of the expression: the radius of gyration of the weights due to the movement of the axis of oscillation, the virtual inertia and the damping factor. This period will not be constant since the values of these components will vary with the ahead speed and with state of sea. The components mentioned probably do not vary independently.

I should mention in passing that the heaving period should be clarified on exactly parallel lines. There is much in common in the approaches to pitching and heaving.

Continuing the discussion on pitching the damping factor is likely to be of greater significance in (c) above than in (b). We know that the linear damping coefficient in rolling increases appreciably with ahead motion. It is likely to do the same with pitching.

There are two other points which have already been made by Professor Robb. In one case I agree with him and in the other I disagree.

I do not think we can interpret the "Resolution" experiments as meaning that we can almost disregard virtual inertia effects in pitching. The value

of I_{yy} will be great. We must not overestimate it, however. For example, if the practical period is half as great again as the basic period, this could be explained by an I_{yy} value equal to 125 percent of the I_y value if other influences are neglected. It is not difficult to envisage a reduction to 80 or 90 percent, however, if damping and change of oscillation are taken into account.

The other point concerns the dynamic magnification factor. We badly need a simple empirical expression for the static angle of repose of a ship in a sinusoidal wave system. This static angle has been explored by Mr. J. L. Kent in his 1922 and 1934 I.N.A. papers, and the expressions deduced are rather complicated. A valuable piece of work will be done by anyone who can produce a simple expression which will give nearly the same results. For a given ship the angle of repose is probably a function of the maximum wave slope and of the ratio of wavelength to ship length only.

5. REAR ADMIRAL C. D. WHEELOCK, U.S.N.:

I want to give my own reactions to some of the tests I witnessed at the David Taylor Model Basin two or three months ago. I think that perhaps they do not reveal erudite theoretical consideration, but they apply to the subject rather directly. When the small models were towed, as has been indicated, a constant towing force was applied and the wavelengths were varied. I think the wave heights were considerably too great to be of real value in any analysis. The author recommends a series of tests with waves half the height of those already used. I think it is probable that the whole range of conditions will have to be exploited until the model we can simulate approximately what happens to the ship at sea. So far as I am aware, we have not yet achieved a case where the conditions result in model behavior corresponding with its natural period in a series of waves.

It is very interesting to notice in these tests that, for wavelengths up to nearly three-quarters of the length of the ship, the model pursued its course without noticeable movement; that is, without pitching and heaving. In waves longer than three-quarter model length, the model pitched in synchronism with the waves, but appeared to be forced about 180 deg. out of phase with the waves; i.e., when the wave crest was at the bow of the model the bow was at its maximum down position. The tests were not pursued to wavelengths beyond about one-and-a-quarter times the length of the ship. It appeared to me that the wave amplitudes were so great as to force the model out of its own period into the wave period in all the tests, and I suspect that it will be necessary in further tests to reduce the wave heights to something very small indeed, in order to see whether we can duplicate what is the experience of all of us at sea, where a ship builds up in pitching to a certain point, then steadies and builds up again, similar to the effect often experienced in rolling.

I am led to wonder whether most of the confusion that comes to the mind of the observer at sea is not due to the smaller waves, having all the appearance of roughness, simply covering up and hiding the waves that are really affecting the ship. During the crossing we have just made in the SS. *America* there was no time at which anyone could detect the waves which were influencing the ship in pitch, because the periods were too long; even on a fairly calm day the surface was so cut up by

short waves that one could not detect the long waves that could possibly be influencing the ship. Only in the Pacific do you find long waves that you can see; I have seen them on several occasions, waves longer than the length of the ship, of 600 to 700 ft., the surface being otherwise so smooth that they can be easily observed.

All those elements, and this matter of virtual mass, are so affected by speed, ship form and other things that it is difficult to present them in any usual form of calculation.

This matter of sea-keeping is tremendously important, and the particular work described in the paper arose from the need for ships of the Navy to maintain speed in rougher and rougher sea. We do not know whether the things we have done are of real value. We do not know, for instance, where by increasing the height of the bow a few feet we can keep a ship at sea in even one force more of sea conditions. We must find that out, and all this work is in pursuance of that end. One of the reasons why this particular kind of test was made, wherein the towing pull was constant, was to determine whether, with a given horsepower or a given thrust, we could achieve higher speed in rougher and rougher waves.

6. PROFESSOR E. V. TELFER (Member of Council, I.N.A.):

My principal comments concern Fig. 2 of the paper. Could the author tell us with what accuracy the spots shown on the curves in this diagram were actually obtained? The spots seem to be quite erratic; and before drawing any lessons from the tests we should be assured that the tests are worth drawing lessons from! For example, it is rather significant that the two best forms, E and D, are not those which have been reported on in practice as being the most seakindly ships. One could too hastily conclude from this that the models failed to interpret the ships, and were therefore useless. Such a conclusion, however, would be erroneous since it is fairly certain that the ships are being reported on in terms of other qualities, possibly dryness, than those shown by the model tests. These latter clearly show low speed loss in heavy weather but if the ships are dirty and ship a large amount of water the captains will slow down speed and thus obscure and lose the intrinsic virtue of the hulls. The lesson here is clearly for the naval architect to re-design the upper structure to secure adequate dryness, and thus allow the captains safely to drive their vessels at the higher speeds for which the hulls are capable.

In discussing speed loss the author shows that whether the initial speed is 20 or 30 knots the effect of adverse weather is to cause roughly about the same loss in speed. While this is true it quite fails to emphasize that the lower speed vessel loses much more speed relatively than does the faster vessel. Viewed from this standpoint, one appreciates immediately the importance of adequate power with which to face a particular weather; and in my opinion this problem is of far greater practical importance than that of the study of the ship motions advocated by the author and others. In other words, a ship should always have sufficient power and size to make the motions induced by the weather relatively unimportant. This may be a counsel of perfection, but I am quite sure that it is clearly the lesson of experience.

There is one point in connection with the author's use of speed loss as a criterion which should be considered. A

form which in fine weather has to expend an inordinate amount of power to achieve 30 knots will, because of its consequent power reserve at lower speeds, be better able to maintain 25 knots than would a vessel which could achieve 30 knots much more easily in fine weather and consequently had her original maximum power correspondingly reduced. The issue here is intriguing, and shows one of the many difficulties of the designer's problems.

7. DR. KENNETH S. M. DAVIDSON (Member of ASNE):

This paper is significant for the broad sweep of its treatment of a subject that is becoming more and more important with the growing urgency of greater speeds in both merchant and naval vessels. In particular, the paper focuses attention on certain points of view that have been gradually gaining headway in recent years.

Foremost among these is the concept that progress toward a truer understanding of ship's motions and of seagoing behavior depends heavily upon a fuller development of the analytical approach. The subject is too complex to hope for big advances from empirical testing alone. Reference (1) of the paper reviews the current status of the analytical approach to seagoing behavior, and the paper itself confirms that various analytical conclusions are well borne out by experiment.

Second is the concept that there may be large differences between the seagoing behavior in irregular seas as compared with the behavior in regular waves, and that these differences need fuller study than they have yet received. The customary assumption of regular, trochoidal waves is an appealing one from the point of view of simplicity. However, the evidence has become increasingly clear that on the one hand this assumption may exaggerate such elementary motions as, for instance, rolling and pitching, and that on the other hand it may fail to give satisfactory explanations of such matters as, for instance, slamming and lurching. It is surely an inadequate assumption on which to place complete reliance, at least without further investigation.

Third, is the concept that among reasonably normal ship forms the limiting seagoing qualities are probably much more a matter of the main characteristics of shape and proportions than of detailed refinements of form. To a first order of approximation, this is of course true also of the full resistances in calm water. There is a difference, however, in that for some years the principal effects of the main characteristics of shape on resistance have been appreciated and attention has been focused on refinements of shape, whereas the principal effects of the main characteristics of shape on seagoing behavior are not yet well understood. This difference means in turn that, for the present at least, the emphasis in studies of seagoing behavior can properly be placed on establishing fundamentals to a first order of approximation, rather than on achieving great precision.

There is an obvious complication in departing from the regular trochoidal wave and introducing irregular seas, that is perhaps even more troublesome from the analytical viewpoint than from the experimental. There is reason to believe, however, that this complication may not prove too great a difficulty if "complex" rather than "irregular" seas comprising combinations of several regular trochoidal components of differing specified sizes moving in specified directions with respect to each other. Irregular seas are visualized as random combinations that would have to be dealt

with statistically; in contrast, complex seas could doubtless be dealt with by linear superposition of several components, thus bringing both analytical and experimental studies of seagoing behavior within range of consistent quantitative treatment. The number of combinations of components that would need to be considered in order to explore adequately the principal features of seagoing behavior, is probably much smaller than might at first be imagined. Perhaps as few as three or four might be enough, and the number must obviously be kept to an absolute minimum if a reasonable amount of ground is to be covered in a reasonable time.

It is obviously desirable in the interests of economy of time and effort that agreement be reached among those engaged in analysis and model-testing regarding the combinations to be employed, and some steps looking to this end have been initiated in the United States during the past several years. The relationships suggested in the paper, wave heights equal to $1 \cdot 1\sqrt{\lambda}$ or $0.55\sqrt{\lambda}$, are interesting in this connection. The suggestion that differences of hull form may prove to be more important with the smaller heights than with the large ones is also interesting.

THE EDUCATION OF NAVAL ENGINEERS

CAPTAIN GUY CHADWICK, USN, RET.

THE AUTHOR

A graduate of the Naval Academy in the class of 1921, CAPTAIN GUY CHADWICK served as a general line officer in battleships, cruisers, destroyers until 1937 when appointed for engineering duty while serving in the Bureau of Engineering. He continued to serve during the amalgamation of the Bureau of Engineering with the Bureau of Construction and Repair to form the Bureau of Ships. From 1938 to 1940 he was Secretary-Treasurer of this Society. During World War II he superintended the machinery and electrical installations at the New York Naval Shipyard of three battleships, six aircraft carriers, and supervised the machinery repairs to 3,200 ships, all classes, domestic and foreign. Following the war he was assigned as Material and Logistics Officer to the Atlantic Reserve Fleet Staff during the period when the fleet was organized and preserved. He then served a tour of duty as Planning Officer, Boston Naval Shipyard. Captain Chadwick is now Commanding Officer, U. S. Naval Training Schools, Massachusetts Institute of Technology, where he also holds an appointment from the Institute as Professor of Naval Engineering.

"To substitute acquired commonsense — knowledge and reflection — for the cruder and tardier processes of learning by hard personal experience and mistakes, is, of course, the object of all education; . . ."

Alfred Thayer Mahan,
From Sail to Steam, 1907.

INTRODUCTION

The science of ships, the *raison d'etre* of the Navy, cannot endure with anything less than its due quota of advance in design. The implicit need for naval officers highly qualified in ship design has long been recognized. The Navy Department regularly sponsors postgraduate instruction in naval architecture and marine engineering.

A survey of the manner of instruction raises many questions. What determines the curriculum and its scope? In what ways may a subject be taught, and on what levels? Is the student's mind considered as well as facts from books? How may creative thinking be released? Is a proper relationship established between the taking-in of knowledge and

the giving-out of thought? Can the student digest what he is fed? Will he be inclined toward continuing self-education after graduation? Will he be ready to transit from student life to productive life? Will he then naturally seek the appropriate level of authority and responsibility?

Assignment to duty as Commanding Officer, U. S. Naval Training Schools, Massachusetts Institute of Technology, and an appointment by the Institute as Professor of Naval Engineering, has given the writer opportunity to sample in professional surroundings the flavor of education and to uncover some receipts.

THE RÔLE OF THE NAVAL ENGINEER

In 1940, the Bureau of Ships, Navy Department, was created by consolidating the Bureau of Construction and Repair with the Bureau of Engineering. The resulting amalgamation of the construction officers with the engineers established a joint profession, now designated as "Naval Engineer." The members of the profession are line officers of the Navy selected for exclusive assignment to engineering duty.

The specialty of the Naval Engineer is the warship. The warship is peculiar to the Navy. The component parts of warships and the weapons and devices which they carry may approximate in basic technology Army, Air Force or civil contrivances. But the warship, the assembly-as-a-whole, is unique in its distinguishing naval identification.

The Naval Engineer's calling derives from the concept of the ship-as-a-whole. It is his responsibility to provide and maintain propelled hulls suitable to carry and serve guns, planes, communications, personnel and cargo, singly or in combination. It is his further responsibility to design ships, not as a mere technician in accordance with existing rules, but as an informed engineer able to anticipate future needs in terms of past experience, current knowledge and research. He is a marine architect in the creative sense. He must be able to adapt and compromise between component parts to produce an effective integrated whole.

This is rather a large order. From an educational point of view if from no other an evaluation of the scope of his primary responsibility as well as a clarification of the breadth of his interest is prerequisite to a program for the selection and instruction of the Naval Engineer.

There is a line of demarkation between the ship and that which the ship carries. This line is a significant and convenient reference. The ship as a carrier for airplanes, guns, communication devices, cargo or personnel is the special province of the Naval Engineer. Whatever is carried is his supporting interest in that he must understand the services which the carrier ship provides.

Three years or more of postgraduate study is currently allowed to ground a Naval Engineer in the fundamentals of his profession. The same length of time would allow an equivalent education of, for example, an Electronics, a Nuclear, a Communications or Ordnance Engineer. Curricula are not interchangeable. The Naval Engineer must be conversant with the needs on board ship of the products of these other engineers. Beyond the demands of a reasonable acquaintance, he should guard against stealing time from primary studies for use in tempting background excursions. He must not sacrifice necessary depth for desirable breadth. Therefore, let us not attempt to educate our Naval Engineers as specialists in everything. Other specialists should be trained.

If a valid need exists for subspecialties within the province assigned to Naval Engineers there are three categories, (1) hull, (2) propulsion, (3) services. Of the latter, the principal

service is electrical, including electronic. The other services which may include the piping of compressed air or steam for machines, ventilation and air conditioning for personnel, or weight handling for cargo, are within the grasp of any competent engineer who may be assigned a particular problem. Thus, the appropriate sub-specialties are:

(1) Hull design and construction,

(2) Ship propulsion engineering,

(3) Marine electrical and electronic engineering.

Greater subdivision invites the real disadvantage of excessively narrow scope.

In reviewing the three sub-specialties, the hull design and construction subspecialty is perhaps less closely related to other branches of technology than the remaining two which may draw components from the finished designs of mechanical and electrical engineers. Thus, the instruction of the former may well be in refinement and in depth compared to the instruction of the latter whose need includes breadth of perception and discrimination in selecting, adapting and applying technological developments.

The Naval Engineer is a designing engineer, the intermediary between scientific knowledge and its technological application in ships. He may work toward developing new scientific knowledge not as a designing engineer but as a scientist. He may operate ships as a technician associated with design only to the extent that operating experience may increase his knowledge. He may administer technological enterprises. In all his activities, he must apply scientific principles. But his basic qualification and his highest duty are in design.

Captain Harold E. Saunders, (CC), USN, in a paper distributed in 1940 among officers of the former Construction Corps described several fundamentals of the design process. For present purposes his thoughts may be paraphrased as follows: Since an effective design must accomplish a specific purpose within an environment, the designing engineer should clearly perceive the purpose and critically visualize the environment. He should be well grounded in theory and well versed in the technology and economics of production and maintenance. He should observe his work take physical form and undergo test. Perfection of design is measured by purity of purpose, directness in the accomplishment of purpose, and simplicity of operation. Captain Saunders' remarks illuminate the design process which should be reflected in the education of designing engineers.

THE SELECTION OF POSTGRADUATE STUDENTS IN NAVAL ENGINEERING

The postgraduate education of Naval Engineers starts with the selection of the individuals to be educated. Officers who have developed an interest in Naval Engineering are the preferable choice. It is peculiarly pertinent in modern days of many specialties competing for the attention of naval officers that midshipmen at the Naval Academy, and officers in the period prior to selection for postgraduate education be carefully informed of the rôle of the Naval Engineer and the dimensions of his specialty. Information given early and elaborated as the experience of the individual increases will help him express his preference.

The interest of the applicant should be motivated by his feeling for the ship-as-a-whole. Once he has stated his desire, the final selection is made on his record. Since the Naval Engineering student must be educable to a high degree, academic competence becomes of first importance. Other good qualities are needed, but adequacy of the educable intellect is an essential ingredient of the student and, later, of the practicing Naval Engineer in his capacity for

advancing the science of ships. Educators experienced in the examination of academic records show remarkable accuracy in estimating the student's academic facility and even in the prediction of his future marks.

LIBERATING THE INTELLECT

The success of the process of education depends upon the effectiveness of a joint endeavor by the student and the teacher to unlock the intellect of the student.

Dr. Daniel L. Marsh who recently retired after a long and distinguished career as President of Boston University in a lecture entitled "Keys of the Kingdom of the Mind" delivered before the U. S. Naval War College, Newport, Rhode Island, March 22, 1949, listed several attributes as "Keys." First comes *"Curiosity"* which Dr. Marsh describes as *"the goad and spur and push of all science."* Then *"Ignition,"* an impulse from within to fire the intellect. *"Hard Thinking"* is a key. Dr. Marsh quotes Henry Ford as saying *"Thinking is the hardest work there is, which is the probable reason why so few engage in it."* Another key is *"Memory,"* a box for knowledge. No man may have cognition beyond the capacity of his memory. Last is *"Practical Application"* to give knowledge reality.

Skillful, indeed, should be the fashioning of these keys, a responsible task for him who would unlock the mind.

THE EDUCATIONAL PROCESS

Education may be considered in three parts. First, are *Skills* that result from training; second, *Facts* which comprise our stock of knowledge; and third, the *Power of Reason* by which knowledge is applied. Education proceeds through these phases in succession, but the process must be repeated many times as the level of the student rises. Thus, we find the three parts intermingled throughout the educational process. No part may be neglected.

SKILLS

Skills are the manners of intellectual behavior. When adequately developed, they become a dependable automatic mechanism. Reading, writing and arithmetic, for example, are elementary skills. There are skills at several levels. A high level skill is space visualization, as the three dimensional geometry or the relative motion of machine parts. A skill of first importance with many subdivisions is that of communication. Reading and writing, listening and speaking, graphics, foreign languages are a few simple components.

Robert B. Heilman, speaking at the University of Washington, said: "One of the most fascinating ironies of academic life is the way in which men in one field, even highbrows in one field, are satisfied, even happy, to be lowbrows in another field. It seems comradely, affable and wellbred to know as little as possible in a colleague's field. I know of one well publicized social scientist who would gleefully announce that he never got more than a 'D' in English, and for all I know, felt that this record improved his mastery of social science."

The ability to communicate with precision is obviously essential to the engineer. Communication in language requires the dual process of sending and receiving. If an engineer sends communications beyond his profession, and if he would clarify communications

within his profession, he should use an accepted and understood vocabulary, avoiding the temptation to coin new words when existing words are adequate, eschewing scientific cant and affectations, and striving to increase his effective vocabulary in order to promote rigor or precision of expression. In receiving communications he should cooperate with the sender and learn to follow context, to grasp what the sender means as distinguished from what he desires or expects him to mean. He should understand that the sender is himself an entity inside a separate skin and that therefore a cooperative endeavor is the only method for attuning two separate nervous systems.

Adequate skill in language communication is not the automatic result of high school or freshman courses in composition. A continuing effort by teachers, including teachers of science and technology, to increase the facility and precision of student language expression is needed. Facility in language oils the thinking mechanism. A technical teacher who fosters expression will gain dividends in improved technical work.

Some forward-looking institutions operate "clinics" in language communication that do not offer general programs but diagnose and prescribe for individual needs. Substantial gains have been made rapidly by students under clinical guidance.

Mathematics are a skill. They are also a language, a good example of rigor. Mathematics are essential to technological understanding and the use of technical knowledge. A measure of the successful teaching of mathematics is the ability of a student to formulate physical problems in mathematical terms.

FACTS

Facts are the components of knowledge. Knowledge can be accumulated only by learning facts. Progress is based on known facts which are the heritage of the generations. With them we may start where our predecessors left off.

Unfortunately a student can never expect to learn all the facts. By devoting full time to the process, he might master a minute segment, probably a small fraction of one percent, but he must reserve some time to learn skills and to develop reasoning power for the application of facts.

What facts then may best be taught? We can divide facts into layers and should allow precedence to basic concepts, *Ohm's Law* over later derivations of electrical relationships; *Force equals Mass multiplied by Acceleration* is a relationship which takes teaching precedence over physical laws of less breadth of application. Indeed, memorization of many electrical and physical laws may not be needed if the basic factual relationships are thoroughly grasped. Basic facts must come first; higher derivations and abstractions can follow as time and opportunity permit.

We may not necessarily believe the same facts today that we do tomorrow, and the teacher must not assume omniscience. That would negate or deny future progress. Above all, facts must not be presented as the natural objective of education. They are only material available for assembly by the intellect into new structures.

REASONING POWER

William G. Saltonstall worried in print about the instruction at Exeter: "I wonder whether we give sufficient thought to the unchanging needs: the stimulation of intellectual curiosity, the insistence on sustained effort, the development of critical and independent thinking . . ."

The development of reasoning power is the highest endeavor of education. We have stimulated the student to develop skills, and caused him to accumulate facts, a process incidentally that Carlyle called, "pouring knowledge into passive buckets." The mere transfer of knowledge is not nearly enough; we must increase the student's power of analysis. No more responsible, difficult or hazardous task faces a teacher. It is the one which may be most rewarding.

The secret of teaching reasoning power lies in the relationship between the teacher and the student. The ancient Greeks believed that the man and the boy must walk and talk together. Confusion of mind because the mutual objectives of the teacher and the pupil fail of clarification by the teacher is a common pitfall. Never, as the story goes, should the palest freshman be startled into incoherent forgetfulness by a sudden and brusque inquiry, "What's your name?" The student must be encouraged to express and defend his convictions.

For his part, the teacher must first and above all avoid the method of the witch doctor, that is, to seek prestige by mystery and indirection. In explaining objectives to the student, he may elect to let the student penetrate a little way behind the scenes and observe the methods of instruction. A good teacher is well fitted to expose the hazards and advantages in methods of application of knowledge, and can insist on rigorous thought, rigorously communicated.

The teacher could well say with advantage to the student: When you graduate, you will be called upon to interpret critically and in terms of useful application the formal and theoretical philosophies of naval engineering. Within these limits you must exercise initiative, assume authority and produce results. You will be "design critics" in the definite sense of "one who expresses a reasoned opinion on any matter involving a judgment of its values, an appreciation of its techniques or an interpretation." You will observe the need for conservatism to preserve a sound engineering practice, and you will likewise see the need for liberalism to improve on past practice. You will understand that both conservatives and liberals should be of responsible thought, critical and creative, and of imaginative initiative. You will strive for that independent intellect which leavens professional thought. Have courage, but observe the need for tact. You are here to study, learn, investigate and criticize; not to march.

The proper teacher will discourage gullibility, and constantly will inveigh against the credulous and easily satisfied mind—the trademark of the uneducated. He will explain the hazards of the assured sounding noises issuing from the mouths of the incompetent, the biased, and the dogmatic. He will warn against catch phrases which mislead, "simplicity" if not of purpose, "logic" if confused with neatness and untested in nature. He might also expose the kind of mind that justifies foolish behavior by blaming the instructions of a superior.

A penetrating sense of perception is the hallmark of a creative engineer. It is a sense that springs from a lively and cultivated feeling for geometric and dynamic relations, for fixed and transient material stresses, for the usefulness of the product in terms of need and for the omnipresent human implications. Without the quality of perception, time is lost and objectives are obscured. It is fitting for the teacher to exercise his pupils in engineering perception by introducing problems and hazards in the application of designs. He will lead them to be discriminative in assumptions and discerning in evaluations.

Method is stock in trade for engineers. Of its three ingredients, the first is "what to do," and is the ingredient most apparent during the course of the educational process. Students are con-

tinually told or led to discover what to do. Although the second ingredient "how to get it done" is less apparent, some practice will normally occur for the student. The third ingredient "when to do it" is the most difficult and the most neglected in education. Time and timing are the lubricants of productive engineering. Suitable timing is only derived from adequate planning. Thus, planning and its servant forethought become the essence of method, planning ignites method.

The student should learn to plan in accordance with nature and the outside world. An assumption that self-generated neat philosophies will necessarily apply in nature will often fail because nature does not always know enough to behave in tidy conformance with desire, especially as to timing.

Students might be gently exposed to the importance of associating with high intellects and sampling their conventions. It need not be a stuffy performance if the association is flexible and the heavy touch avoided. The student might learn that the cultivation of a sense of the vulgar can sometimes be a mistaken instinct. The feeling that all the more delightful forms of recreation lie in this direction is a mistaken emotion; a full diet of comic books, detective stories or sensational daily newspapers lacks vitamins.

It could be becoming to the teacher to re-emphasize the difficulties of communication, notably the fact that the teacher may have a background of context at variance with that of the student, and that a high use of perception is needed to establish close rapport. The teacher can offer a special vertical kind of knowledge in a prescribed field; he desires to impart this knowledge to the student, but is unlikely to be persuaded that he should give something else. With patience and persistence, he might help the student to discover the universal value of close personal relationships based on mutual interest supported by good manners.

A common contention by employers is that newly graduated students are of little use, and that it takes considerable time for them to become helpful. Because a student is apt to be governed by rules before reason, if the rules do not fit new surroundings, the student flounders. Success attained in the third part of the educational process, the development of reasoning power, will be measured in the reduction of the period of time between graduation and the day when the student begins to do useful work.

MARKS

Marks are inherent in the educational system, and marking is a difficult process. It imposes hard work on the teacher and may arouse reactions in the student ranging from apathy to pride, or from annoyance to satisfaction. Nevertheless, no adequate educational system has yet been devised without the use of marks. It is usual to avow that their purpose lies in the evaluation of the student, but they can exert a substantial and, sometimes, a controlling influence on the quality of the teaching process. Marks may stimulate the student by arousing his competitive spirit. Desire for the kudos of high marks and the fear of low marks are potent inducements for effort; fear of low marks may dull a student's sensibility and upset his intellectual equilibrium. The direction of emphasis of the marking method is so important in its effects on the student that a need arises for more than a superficial understanding.

The marking method should furnish equitable support to the three parts of education discussed above: skills, facts and the power of reason. A balance of emphasis is requisite. Since marks are

mainly derived from answers by the student to questions posed by the teacher, it is apparent that the nature of these questions together with the manner of their presentation is essential.

Questions are of two types depending upon the kind of answer invited. The first permits no deviation in an appropriate answer and is used in "true and false" examinations. Another example is the mathematical problem whose answer is either right or wrong. This kind of question is valuable in demonstrating the student's knowledge of "facts" and to some extent his "skills" as shown in the manipulation of a formula, or the ability to derive formalized relationships, although such a question can hardly demonstrate his skill in the use of language beyond the requirements of mere grammar. Other advantages lie in the ease with which marks may be computed by the teacher using a kind of accounting procedure. The seductive neatness of the accounting manipulation is apt to be psychologically satisfying to the student. For want of a better term, we may christen the sort of mark thus derived as an *"arithmetical mark."*

Archibald MacLeish warned in these words against a kind of teaching, ". . . indoctrinated rather than instructed, whose neuter and unasking minds packed with creed instead of questions . . . will give to every stimulus the appropriate response."

The selective presentation of facts can depart from the sphere of education to become a method of propaganda indoctrination, of which one branch is examination by specific problem. The second kind of question invites the student to exercise his mind beyond the process of litany, with an opportunity to resist dogma by injecting his own thoughts into the answer. He may demonstrate essential qualities of the intellect, for example, lack of gullibility, critical sense, or courageous imagination. He may display his inner compulsion to extend his thoughts beyond the text. He must exercise his ability to communicate in language. Marking the answer to a question of this type requires judgment. Therefore, let us term this species of mark a *"judgment mark."*

Judgment marks may be less appealing to a student than arithmetical marks because his judgment may not agree with that of the teacher; there is no façade of arithmetic to simulate infallibility. Judgment marks are certainly more onerous for the teacher. One measure of the quality of the teaching process may well be the balance maintained between arithmetical and judgment marks.

THE CURRICULUM

"There is danger of educational indigestion if we spread the student's attention over too many specialties . . . We must concentrate on a thorough mastery of the important basic essentials if we are to educate citizens capable of self-government and self-discipline."

William G. Saltonstall, (Exeter).

The undergraduate curriculum of the Naval Engineer should be as liberal in nature as time will permit, without neglect of the sciences. Whether completed at the Naval Academy or at another institution, a broad educational foundation should be established. Specialism in depth should be reserved for postgraduate education.

Three concepts belong to a curriculum specially oriented for Naval Engineers. The first is the notion of the ship-as-a-whole where each element of design is measured as a contribution in balanced compromise with other elements that may not be displaced. The second is that education is for future use. The achievements of today are not

the aims of tomorrow, and current devices become important only in the illustration of technological fundamentals. The third is that application in nature is the aim of any engineer. His instruction should be well supported by training in the use of science and technology for resolving physical problems.

The categories special to the curriculum of Naval Engineers may be conceived in a series of levels to be filled in succession from the lowest. The first level should comprise a continuous exercise and review in the communication of ideas by language, mathematics and graphics. The second level should present the fundamentals of mechanics, thermodynamics, electricity and nuclear reactions, the characteristics of materials, and basic economics. The third level is a refinement of the second; for example, electricity divides into magnetism, direct and alternating current, and electronics; thermodynamics becomes heat engines, refrigeration, and heat transfer. The fourth level considers the components of the ship; the hull, the propulsion plant and the auxiliaries, together with the control devices including servomechanisms and communications. The fifth level is an example of refined design of a ship component. The final and highest level is the selection of parts, design and assembly of a ship.

THE CONTENT OF THE CURRICULUM

A recent evolutionary development in a 3-year postgraduate curriculum for Naval Engineers carefully preserves the main stem of design from the point of view of the ship-as-a-whole. Grafted upon the main stem are 4 sub-specialties, (1) Hull Design and Construction, (2) Ship Propulsion Engineering, (3) Marine Electrical Engineering and (4) Electronics Engineering. The individual subjects within the curriculum are selected meticulously and arranged in neat array, an imposing package of education. What is the value of its content and how may this be measured?

The teacher is the principal influence on the content of the curriculum. The variety of subjects comprise many teaching specialties. The best teacher devotes his entire time to his specialty, an advantage found in large institutions which can keep each teacher fully employed in one sphere of work.

Many of the subjects are not the sole province of Naval Engineering but may be taught in other engineering areas of an institution devoted to general technological education. In actual practice the classroom emphasis is apt to be swayed by the teacher's particular interest, or that of a large student segment. A small group of Naval Engineers studying in a class which has a majority of students in another specialty and a teacher more conversant with shore than with ship needs may not realize optimum emphasis in the subject matter, and the direction of emphasis can vary from year to year.

Some of the subjects have particularly to do with ships. The teaching staff must have a group of ship specialists in the categories of naval architecture, ship propulsion and ship electrical engineering, not only to teach these specialties but also to develop and coordinate the entire curriculum.

Supporting the teacher are the classroom, lecture hall, laboratory, drafting room and library. Here are found the sources of *knowledge;* and *skills* are exercised. The laboratory and drafting room are the especial abode of *reasoning power.* Fortunate indeed is the student who in his final semesters becomes associated with a teacher, exploring the threshold of new knowledge in the laboratory of that teacher.

The amount of time allocated to a course modifies the content of the curriculum. Three years of postgraduate study appear to be the minimum to edu-

cate a designing engineer in the fundamentals. Even then, there are substantial omissions.

For some students it is evident that the end of a 3-year period marks a point where education progresses at an accelerated pace. A fourth year of study for a few doctoral candidates is amply justified in terms of investment by the Navy.

Continuity influences the content of the curriculum. Some subjects are prerequisites for others. The highest endeavor of the last year is in the original work termed *thesis,* properly a joint undertaking of student and teacher in which the student bears primary responsibility for developing new knowledge, and the teacher acts as advisory consultant. In preparation for the thesis the student not only needs familiar acquaintance with the staff and the equipment of the institution, but also benefits by formal prerequisite studies cut to suit the local environment of the thesis. The advantage becomes more apparent for theses in advanced areas inhabited by outstanding teachers. It is preferable for coordination that the entire postgraduate curriculum be undertaken at the same institution.

The sequence of the content of the postgraduate curriculum is first, basic engineering, next, formal studies in support of the Naval Engineering specialty, and last, the thesis atmosphere of original research. The thesis gives the student a preview of his professional life. It gives him practice in combining study with creative endeavor and demonstrates the natural interchange between education and design. It fits him for future self-education, and fosters his desire thus to support his career as a Naval Engineer. The propriety of an advanced curriculum may be judged by the amount of time devoted to the thesis where a neat listing of "subjects" is not consonant with the originality of the work.

CONCLUSION

The quality of the education of Naval Engineers is a matter of high import to the operating officers of the Navy. It is not enough to teach the formalities of existing practice. Students who can learn traits of intellectual behavior that will foster improvement in future ship design must be selected. The development of these traits is strongly influenced by educational method. The manner of education is susceptible of critical analysis. The suitability of a curriculum may be determined in terms of explicit educational processes.

APPENDIX I

A Contemporary Example in Four Subspecialties of Postgraduate Curricula for Naval Engineers.

Curricula at the Massachusetts Institute of Technology leading to the Degree of Naval Engineer for Naval Officer Students in Naval Construction and Engineering, Course XIII-A.

Note: The columns of three figures denote hours per week scheduled for the student. The first column is lecture time. The second column is laboratory or drafting room time. The third column is preparation time.

I

Hull Design and Construction
Course XIII-A-1

Summer 1951

2.046	Strength of Materials	4–2–8
8.03	Physics (Electricity)	4–1–5
13.20	Elementary Ship Design	1–4–0
M73	Review of Mathematics	20–0–0
		49

Term I (Fall 1951)

2.081	Strength of Materials	3–0–6
2.402	Heat Engineering	4–2–7
3.391	Properties of Metals	2–1–3
10.311	Heat Transfer	3–0–3
13.17	History of Warships	3–0–4
M351	Advanced Calculus for Engineers	3–0–6
		50

Term II (Spring 1952)

1.40	Structures	3–3–6
1.612	Fluid Mechanics	3–0–6
13.01	Statics of Ship Des.	3–1–5
13.12	Warship General Arrangements	2–0–4
13.21	Warship Form Design	0–5–0
M352	Advanced Calculus for Engineers	3–0–6
		50

Term III (Fall 1952)

1.63T	Applied Hydromechanics	3–0–6
6.181	Electrical Engineering	3–1–6
13.02	Dynamics of Ship Des.	2–2–4
13.13	Warship Structure, Theory I	3–0–6
13.22	Warship General Design	0–5–0
13.75	Warship Propulsion I	3–2–6
		52

Term IV (Spring 1953)

1.42	Structures	3–0–6
1.683	Experimental Hydromechanics	0–6–0
3.392	Properties of Metals	2–1–3
6.191	Electrical Engin.	3–1–6
13.14	Warship Structure, Theory II	3–0–6
13.24	Warship Structure, Design II	0–5–0
13.76	Warship Propulsion II	2–4–4
		55

Term V (Fall 1953)

2.671	Air Conditioning	3–0–9
6.351	Engineering Acoustics	2–2–5
13.15	Warship Basic Design I	3–0–6
13.16	Warship Basic Design II	3–0–6
13.25	Warship Structure, Design II	0–5–0
13.54	Marine Engineering Dynamics	2–0–4
	Thesis	0–3–0
		53

Term VI (Spring 1954)

1.58	Elastic Stability of Flat Plates	3– 0–6
2.126	Experimental Stress Analysis	2– 3–4
13.26	Preliminary Design of Warships	0– 9–0
14.113	Economics & Labor Relations	4– 0–8
	Thesis	0–17–0
		56

II

Ship Propulsion Engineering
Course XIII-A-4

Summer 1951

2.046	Strength of Materials	4–2–8
8.03	Physics (Electricity)	4–1–5
13.20	Elementary Ship Design	1–4–0
M73	Review of Mathematics	20–0–0
		49

Term I (Fall 1951)

2.081	Strength of Materials	3–0–6
2.402	Heat Engineering	4–2–7
3.391	Properties of Metals	2–1–3
10.311	Heat Transfer	3–0–3
13.17	History of Warships	3–0–4
M351	Advanced Calculus for Engineers	3–0–6
		50

Term II (Spring 1952)

2.422	Heat Engineering	2–0–4
3.392	Properties of Metals	2–1–3
13.01	Statics of Ship Des.	3–0–6
13.12	Warship General Arrangements	2–0–4
13.21	Warship Form Design	0–5–0
14.113	Economics & Labor Relations	4–0–8
M352	Advanced Calculus for Engineers	3–0–6
		53

Term III (Fall 1952)

2.49	Fluid Mechanics, Adv.	3–0–9
6.181	Electrical Engineering	3–1–6
13.02	Dynamics of Ship Des.	2–2–4
13.13	Warship Structure, Theory I	3–0–6
13.22	Warship General Design	0–5–0
13.75	Warship Propulsion I	3–2–6
		55

Term IV (Spring 1953)

2.261	Lubrication & Friction	2–0–4
2.491	Compressible Fluid Mechanics	3–0–9
2.783	Control Problems	3–0–9
6.191	Electrical Engin.	3–1–6
13.241	Warship Structure, Design I-A	0–2–0
13.76	Warship Propulsion II	2–4–4
		52

Term V (Fall 1953)

2.213	Gas Turbines I	3–0–9
2.79	Internal Combustion Engines, Elementary	3–0–6
13.15	Warship Basic Des. I	3–0–6
13.16	Warship Basic Des. II	3–0–6
13.73	Mechanical Vibration	3–0–6
13.79	Propeller Design	2–1–2
	Thesis	0–3–0
		56

Term VI (Spring 1954)

2.28	Fluid Machinery	3– 0–9
13.26	Preliminary Design of Warships	0– 9–0
13.74	Mechanical Vibration	3– 0–6
	Thesis	0–17–0
		47

III

Marine Electrical Engineering
Course XIII-A-2

Summer 1951

2.046	Strength of Materials	4–2–8
8.03	Physics (Electricity)	4–1–5
13.20	Elementary Ship Design	1–4–0
M73	Review of Mathematics	20–0–0
		49

Term I (Fall 1951)

2.081	Strength of Materials	3–0–6
2.402	Heat Engineering	4–2–7
3.391	Properties of Metals	2–1–3
10.311	Heat Transfer	3–0–3
13.17	History of Warships	3–0–4
M351	Advanced Calculus for Engineers	3–0–6
		50

Term II (Spring 1952)

1.612	Fluid Mechanics	3–0–6
6.00	Electrical Engineering Principles	4–0–8
13.01	Statics of Ship Des.	3–1–5
13.12	Warship General Arrangements	2–0–4
13.21	Warship Form Design	0–5–0
M352	Advanced Calculus for Engineers	3–0–6
		50

Term III (Fall 1952)

6.01	Electrical Engineering Principles	3–0–6
6.75	Electrical Measurement Laboratory	3–0–6
13.02	Dynamics of Ship Des.	2–2–4
13.13	Warship Structure, Theory I	3–0–6
13.22	Warship General Design	0–5–0
13.75	Warship Propulsion I	3–2–6
		51

Term IV (Spring 1953)

3.392	Properties of Metals	2–1–3
6.02	Electrical Engineering (Electronics)	3–0–6
6.031	Electrical Engineering Principles	2–1–6
6.76	Electronics Laboratory	1–2–3
13.241	Warship Structure, Design I-A	0–2–0
13.76	Warship Propulsion II	2–4–4
14.113	Economics & Labor Relations	4–0–8
		54

Term V (Fall 1953)

6.032	Electrical Engineering Principles	3–0–6
6.20	Electronic Control & Measurement	3–0–6
6.605	Servomechanisms	3–0–6
13.15	Warship Basic Des. I	3–0–6
13.16	Warship Basic Des. II	3–0–6
13.54	Marine Eng. Dynamics	2–0–4
	Thesis	0–3–0
		54

Term VI (Spring 1954)

6.606	Servomechanisms	3– 3–6
6.86	Electrical Machinery Laboratory	2– 4–4
13.26	Preliminary Design of Warships	0– 9–0
13.70	Warship Electrical Engineering	2– 2–4
	Thesis	0–17–0
		56

IV

Electronics Engineering
Course XIII-A-3

Summer 1951

2.046	Strength of Materials	4–2–8
8.03	Physics (Electricity)	4–1–5
13.20	Elementary Ship Design	1–4–0
M73	Review of Mathematics	20–0–0
		49

Term I (Fall 1951)

2.081	Strength of Materials	3–0–6
2.402	Heat Engineering	4–2–7
6.003	Electrical Engineering Principles	3–0–6
13.17	History of Warships	3–0–4
10.311	Heat Transfer	3–0–3
M351	Advanced Calculus for Engineers	3–0–6
		53

Term II (Spring 1952)

1.612	Fluid Mechanics	3–0–6
6.02	Engineering Electronics	3–0–6
13.01	Statics of Ship Des.	3–0–6
13.12	Warship General Arrangements	2–0–4
13.21	Warship Form Design	0–5–0
M352	Advanced Calculus for Engineers	3–0–6
		47

Term III (Fall 1952)

3.391	Properties of Metals	2–1–3
6.20	Electrical Control & Measurement	3–0–6
6.756	Electrical Measurement Laboratory	0–4–4
13.02	Dynamics of Ship Des.	2–2–4
13.13	Warship Structure, Theory I	3–0–6
13.22	Warship General Design	0–5–0
13.75	Warship Propulsion I	3–2–6
		56

Term IV (Spring 1953)

3.392	Properties of Metals	2–1–3
6.321	Electrical Communication Principles	3–0–6
6.561	Network Theory, Adv.	3–0–6
6.581	Transients in Linear Systems	3–0–6
6.80	Electrical Measurement Laboratory	0–4–4
13.241	Warship Structure, Design I-A	0–2–0
13.76	Warship Propulsion II	2–4–4
		53

Term V (Fall 1953)

6.291	Principles of Radar	3–0–6
6.34	Electrical Communication Laboratory	0–3–3
6.621	Microwave Circuit	3–0–6
13.15	Warship Basic Des. I	3–0–6
13.16	Warship Basic Des. II	3–0–6
13.54	Marine Eng. Dynamics	2–0–4
	Thesis	0–3–0
		51

Term VI (Spring 1954)

6.605	Servomechanisms	3– 0–6
6.222	Antennas	3– 0–6
13.26	Preliminary Design of Warships	0– 9–0
14.113	Economics & Labor Relations	4– 0–8
	Thesis	0–17–0
		56

SEMI-ACCELERATED CORROSION TESTS OF MEDIUM AND HIGH TENSILE STEEL IN CONTACT WITH PHENOLIC FOAM

H. J. STARK, J. KAMINETSKY AND R. R. WINANS

The Authors

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He is a member of American Society of Naval Engineers, Society of Plastic Engineers, Association of Senior Engineers of the Bureau of Ships, Alpha Chi Sigma, Kappa Kappa Psi, Horological Institute of America, Washington Rubber Group, and Reserve Officers Association of America.

R. R. Winans graduated from the Newark College of Engineering in Newark, New Jersey in 1928 with a degree of Bachelor of Science in Electrical Engineering. He did his cooperative work at the Bakelite Labs in Bloomfield, New Jersey and was employed there for over a year after graduating. After several years in various Radio and Radio Tube Manufacturing plants he was employed by the Material Laboratory, New York Naval Shipyard in 1935 as a Materials Engineer.

In 1943 he was made head of a newly formed Section on Electrical Insulating Materials. For the past several years he has been involved mainly with Rigid Plastics and is in charge of all of the work in the Material Laboratory on this subject.

He is a member of the American Society for Testing Materials, Committee D-9 on Electrical Insulating Materials, Committee D-20 on Plastics and the Conference on Electrical Insulation of the National Research Council.

Julius Kaminetsky received the degree of B.S. in Chemistry at Brooklyn College in 1936. Currently is pursuing graduate work at Brooklyn Polytechnic Institute in technology of plastics and resins. Worked as chemist until the beginning of the war in the varnish industry. Appointed as chemist in the Plastics section of the Chemical Warfare Service in 1942, visiting molding and fabricating plants where he tested materials purchased on government contracts. Did research on Napalm (gasoline thickener) for incendiary bombs and on cellulosics. In charge of inspection of plastic lenses and gas mask parts. From 1945 to 1947 was employed by the War Assets Administration as technical classifier of surplus chemicals, paints, petroleum products, and plastics. From 1947 to present, employed as Technologist in the Rigid Plastics Section of the New York Naval Shipyard, on developmental and research work on the effect of elevated temperatures and water immersion on plastic materials. For the past year, worked on evaluation of low density cellular materials for Naval applications.

CORROSION TESTS

Introduction

The utilization of Phenolic Foamed-In-Place type of resin where it is in continuous contact with steel and subject to alternate wet (salt water) and hot dry conditions as might be encountered in Naval applications raises the question concerning a possible corrosive effect of the Phenolic foam on the steel. Knowledge concerning this fact is important because it determines whether steel surfaces in contact with these materials must be protected from corrosion.

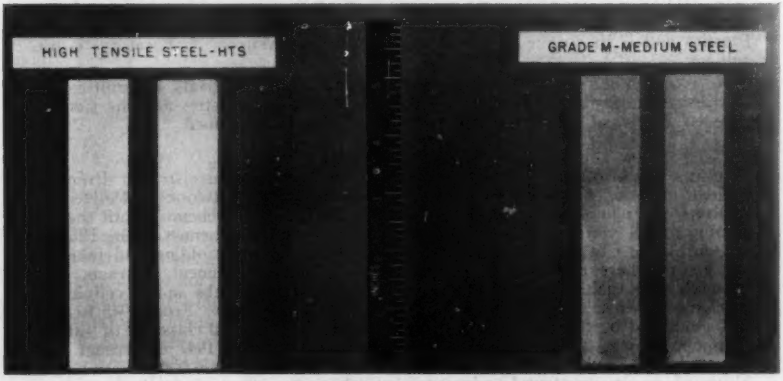
Objective

The objective of this work was to determine, under the conditions of test, whether the materials described herein contributed to or inhibited the corrosion of steel and the relationship if any between the pH measurements on water extracts of crushed samples and the observed corrosive effect.

Materials

1. Phenolic Foam B
2. Phenolic Foam C
3. Modified Phenolic Foam C
4. Balsa Wood

Materials 1 and 2 are commercially obtainable products. They were mixed according to recommended procedure. Material 3 has been modified by the addition of 1% of another commercially available resin, and was mixed for 25 minutes before addition of the acid ingredient and pouring into the mold. The balsa wood sample was obtained from Naval Shipyard stock. Proprietary names of these materials are not used for reasons that are obvious to the readers.



Appearance of Steel Specimens
(Before Immersion)
Illustration 1

TABLE I

Grade M—Medium Steel

Steel Specimen Number	Actual Weight Loss (gms)	% Weight Loss	Size of Specimen L.	W.	T.	Density	Material Attached to Steel
21	.3294	.41	–	–	–	—	Control
22	.3161	.39	–	–	–	—	Control
23	.4466	.55	5	2	2	1.6	Phenolic Foam B
24	.4259	.53	5	2	2	1.6	Phenolic Foam B
25	.4475	.55	5	2	1	2.9	Phenolic Foam B
26	.5037	.60	5	2	1	3.0	Phenolic Foam B
27	.5365	.66	5	2	1	5.3	Phenolic Foam B
28	.4660	.56	5	2	1	5.3	Phenolic Foam B
29	.5125	.61	–	–	–	—	Control
30	.4663	.55	–	–	–	—	Control
31	.9226	1.09	5	2	1	2.17	Phenolic Foam C
32	.8295	.98	5	2	1	2.28	Phenolic Foam C
33	.8103	.96	5	2	1	2.40	Modified Phenolic Foam C
34	.8262	.99	5	2	1	2.40	Modified Phenolic Foam C
35	.7120	.87	5	2	1	9.25	Balsa Wood
36	.7548	.89	5	2	1	8.94	Balsa Wood

METHOD OF TEST

Specimens of steel, with and without foam or wood attached were immersed in synthetic sea water for 12 hours at room temperature (77° F) and then dried for 12 hours at 90°F in a circulating air oven. Illustration 6 shows the appearance of Phenolic foam C (top) and Phenolic foam B (bottom) and Method of Attaching Specimen before test. This alternate wet and dry condition continued for 14 consecutive days. The steel specimens were weighed prior to and after conditioning for 14 days, making certain to remove the products of corrosion with ammonium citrate before the second weighing. The loss in weight as a result of this cycling is determined and reported. This weight loss is considered as due to corrosion.

Steel samples were sand blasted before test. The Phenolic foam samples had no skin or rind on their surface. The steel specimens used were 5″ by 1″ by ⅛″ and weighed approximately 85 grams each. Two grades of steel were used; HTS high tensile steel and grade M medium steel. Phenolic foam and balsa wood specimens were cut to 5″ by 2″ by 2″ and in some instances 5″ by 2″ by 1″. Specimens of Phenolic foam and balsa wood were attached to the steel specimens so that 2½ inches

TABLE II

*High Tensile Steel—HT—(48-S-5)**

Steel Specimen Number	Actual Weight Loss (gms)	% Weight Loss	Size of Specimen L.	W.	T.	Density	Material Attached to Steel
1	.2796	.33	–	–	–	—	Control
2	.2827	.33	–	–	–	—	Control
3	.3687	.44	5	2	2	1.6	Phenolic Foam B
4	.4031	.48	5	2	2	1.6	Phenolic Foam B
5	.4183	.49	5	2	1	2.8	Phenolic Foam B
6	.3988	.47	5	2	1	2.8	Phenolic Foam B
7	.4466	.53	5	2	1	5.2	Phenolic Foam B
8	.4422	.53	5	2	1	5.2	Phenolic Foam B
9	.3113	.37	–	–	–	—	Control
10	.3401	.40	–	–	–	—	Control
11	.7185	.85	5	2	1	1.98	Phenolic Foam C
12	.7788	.92	5	2	1	2.06	Phenolic Foam C
13	.8048	.95	5	2	1	2.40	Modified Phenolic Foam C
14	.6389	.77	5	2	1	2.47	Modified Phenolic Foam C
15	.7574	.90	5	2	1	7.61	Balsa Wood
16	.7326	.87	5	2	1	8.07	Balsa Wood

* Navy Dept. Specification, Steel, Plate, Hull and Ordnance, Structural, Black (Uncoated) and Zinc coated (Galvanized).

of the metal specimen overhung the test foam specimen on one end. See illustration 6. The wide face of Phenolic foam and balsa wood material was in direct contact with the wide face of the steel specimen. Fibrous glass tape was used to fasten the metal to the foam or wood specimens. Duplicate specimens were used in the determinations.

pH determinations were made on a 1% mixture (one gram of foam or wood in 99 c.c of distilled water). The crushed foam (or wood) was allowed to remain soaking in water for 48 to 72 hours.

DISCUSSION OF RESULTS

Tables I and II report the actual weight loss and the percent weight loss for the various metal specimens as a result of contact with the Phenolic foam and balsa wood of respective density. Control samples of steel were used in all cases. The test is considered in the nature of a semi-accelerated corrosion test. It was selected so as to have a comparatively simple and easily reproducible set of conditions. Corrosion of the metal control samples in all cases occurred. Inasmuch as the foams produced an acid medium in synthetic sea

TABLE III

pH Values

Material	*Density*	*pH Value*
Phenolic Foam B	1.6	2.2
Phenolic Foam B	1.6	2.2
Phenolic Foam B	2.8	2.1
Phenolic Foam B	2.7	2.2
Phenolic Foam B	5.3	2.2
Phenolic Foam B	5.3	2.2
Phenolic Foam C	2.7	4.0
Phenolic Foam C	2.5	4.5
Modified Phenolic Foam C	2.5	3.5
Modified Phenolic Foam C	2.5	3.5
Balsa Wood	8.8	5.8
Balsa Wood	8.2	5.1

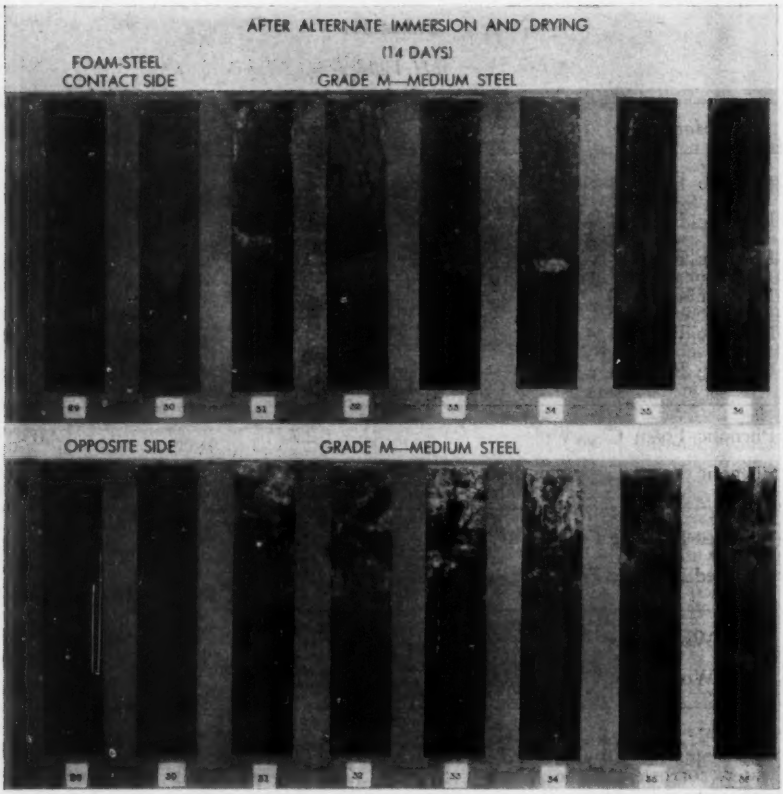
water, corrosion of the steel was to be expected.

The percent loss in weight due to corrosion was greater in the steel specimens that were in contact with the foam and balsa wood than those that had no foam or balsa wood attached. This can be observed in the data reported in Tables I and II and also specimens in Illustrations 2, 3, 4, and 5. The percent weight loss figures for the medium steel as a result of this test appear to be slightly higher than for the high tensile steel.

In the case of Phenolic foam B the degree of corrosion increased with the density of the foam specimens.

The degree of corrosion caused by the modified Phenolic foam C was no greater than that produced by the unmodified Phenolic foam C. The corrosion due to the presences of either Phenolic foam C or modified Phenolic foam C, however, was greater than that produced by Phenolic foam B. The balsa wood specimens produced as much corrosion as did the Phenolic foam C and modified Phenolic foam C specimens but more than Phenolic foam B.

The influence or effect of pH on corrosion is an interesting observation. The actual values obtained are reported in Table III. Phenolic foam B gave the lowest pH values. It is significant to

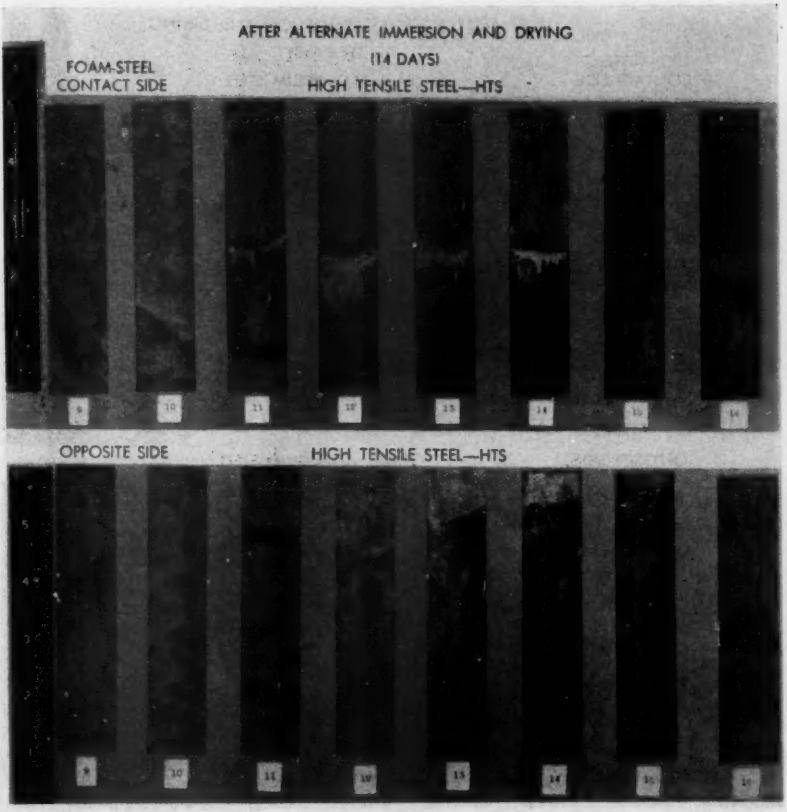


Phenolic Foam C
Modified Phenolic Foam C
Balsa Wood
Illustration 2

note balsa wood gave an acid pH. From the observed data, the pH values of Phenolic foam C specimens were higher than for the modified Phenolic foam C specimens. Both Phenolic foam C and modified Phenolic foam C materials produced higher pH values than did the Phenolic foam B specimens.

Although the Phenolic foam B specimens in the density tested were more acidic (lower pH readings), the corrosive effect produced by their contact with the steel specimens was less than that produced by the Phenolic foam C and balsa wood specimens.

From the observed result it appears that under the conditions of tests and with the materials tested there does not follow any correlation of pH values with the corrosive effect observed.



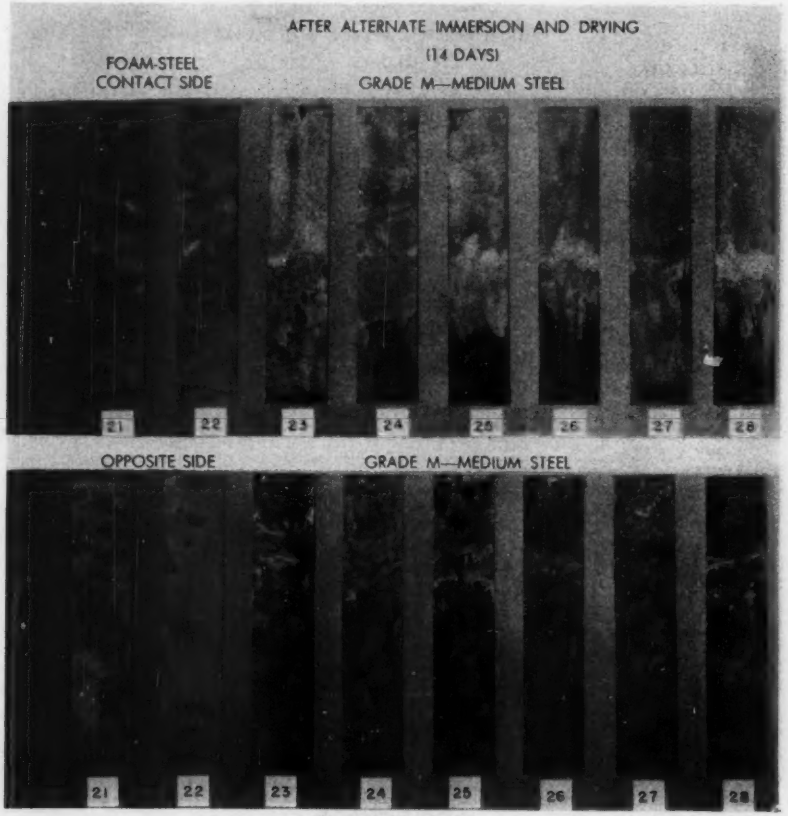
Phenolic Foam C
Modified Phenolic Foam C
Balsa Wood
Illustration 3

CONCLUSIONS

Both Phenolic foams of the type described and balsa wood when in direct contact with steel and subjected to alternate synthetic salt water immersion and hot and dry conditions, produce a slightly accelerated corrosion in high tensile and mild steel.

The corrosive effect of the Phenolic foam C and modified Phenolic foam C and balsa wood are of the same order; however, the corrosive effect of the balsa wood is greater than that of Phenolic foam B.

There does not appear to be any correlation of pH values with the corrosive effect observed.



Phenolic Foam B
Illustration 4

The degree of corrosion increased with the density of Phenolic foam B material.

pH values do not appear to vary with the density of Phenolic foam tested.

Where applications involving the use of these materials is encountered and conditions such as those described herein are possible or existent, it is recommended that the steel surfaces in contact with these materials be protected by suitable coatings.

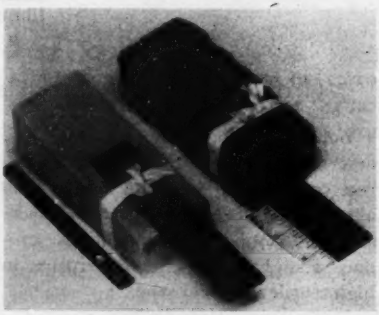
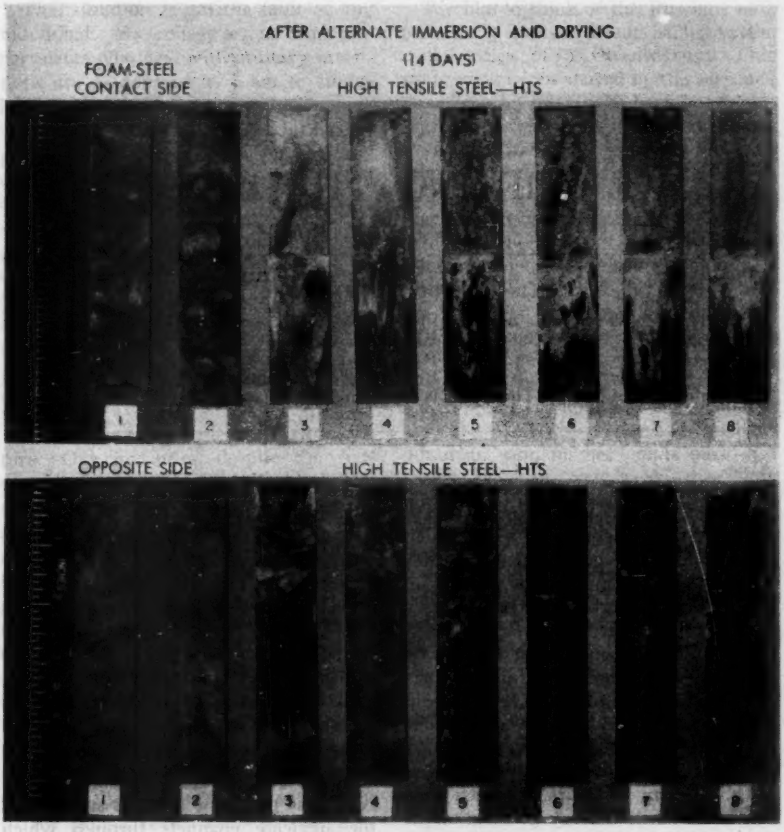


Illustration 6



Phenolic Foam B
Illustration 5

ACKNOWLEDGMENT

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ARE WE STIFLING THE INVENTORS?

ACKNOWLEDGMENT

This article is reprinted by special permission of "The Saturday Evening Post," copyright 1951 by the Curtis Publishing Company. It appeared in the June 9, 1951, edition of that magazine. The authors are CAPTAIN GEORGE N. ROBILLARD, USN, Assistant Chief of Naval Research for Patents, and BEVERLY SMITH, Washington Editor of "The Saturday Evening Post."

America must invent or die. It is as simple as that. We and our allies are engaged in a struggle, the end of which no man can foresee, against the communist powers dominated by the Soviet Union. To counterbalance their greater manpower we have, for the time being, technical superiority in land, sea, air and atomic weapons. So far, this has deterred the men in the Kremlin from attempting a world-wide war of conquest. If our technical superiority is ever lost or even seriously narrowed down, we may find ourselves in a conflict which will destroy civilization as we know it.

If, on the other hand, we can widen our technical margin in weapons, meanwhile maintaining adequate armed forces to handle the finished product, the prospects for peace will brighten. The Soviet leaders are not likely to start a world war unless they think they can win.

What then has America to fear? Aren't we renowned for our inventiveness? Don't we have the greatest research laboratories in the world, the largest number of trained scientists, engineers, technicians and production men? Undoubtedly so.

Yet there are flaws at the heart of our inventive system which, unless corrected, may prove fatal. The flaws have not attracted the attention of the President, or the Congress, or the general public, because they are obscure, insidious flaws, like a vitamin deficiency which covertly weakens an apparently well-nourished man. These flaws concern our system of stimulating, evaluating, testing and rewarding inventions primarily useful for military purposes.

My work as patent counsel for the Navy has given me an intimate view of the intricate channels through which military inventions flow—or are supposed to flow—from the mind of the inventor into eventual acceptance and use by the armed forces. This study has forced me to the following reluctant conclusions: (1) the stimulus, the incentive, to make such inventions is weak, especially as it affects the independent inventor or the inventor employed by the Government; (2) methods of evaluating and testing such inventions are inadequate; (3) the rewards for such inventions are usually capricious or paltry; and (4) the situation has grown worse rather than better in the last sixteen months.

This situation is not the fault of any individual. My colleagues in other departments who deal with military inventions are as anxious as I am to stimulate, develop and reward valuable new ideas. But we are all caught in a complex of laws, regulations, orders and customs. Many of these barriers were originally designed to protect the Government against the occasional greedy and unscrupulous inventor, but they also serve to discourage the great majority of inventors who are honest and patriotic. Uncle Sam is now so well protected against inventors that he may well miss out on the very inventions which could save his neck.

I don't want to paint too dark a picture. Our inventors, despite the red-tape entanglements and legal booby traps in the path, are continuing to devise a fair number of valuable new military items. Some of these men go broke; some—especially if they have strong corporate backing—are suitably rewarded. In Government laboratories many gifted technicians, on modest salaries, are devoting to military invention a talent which, if employed in the industrial world, would bring them big money. They continue to do this, although the Government has recently curtailed their chance of ever sharing in the profits which their patents may later yield commercially. But some of these men, I know, are unhappy and discouraged under the new restrictions —a state of mind not conducive to the best results.

Yes, in spite of all the difficulties, Americans are still coming up with some new military ideas. Out of the vast reservoir of American ingenuity and inventiveness, we are getting at least a trickle devoted to the little matter of national survival. Even with this trickle, we may be outinventing the Russians. I hope so, but we cannot be sure.

We like to think of the Russians as a technically backward folk, baffled by the complexities of a wheelbarrow. This smug view is not shared by the seasoned American jet pilots who have been dueling with Russian MIG-15 jet planes over the Yalu River. So far, we have more than held our own with the MIG's, but news reports from Korea indicate that this is not because our jets are better, but because our superbly trained pilots are superior to the anonymous airmen—we don't even know whether they are Russians or Chinese—who fly the MIG's. Some of our pilots assert that the MIG is actually a better and faster plane than ours. Gen. Carl Spaatz, USAF (Ret.), who knows his airplanes, says the MIG-15 is "about as good as our own," and further points out—in a recent issue of Newsweek—that "any nation which can produce as good a fighter as the MIG-15 also is capable of producing a first-class bomber, and sooner or later will do so."

The deliverability of atomic bombs depends largely on airplane capabilities. Our margin here has been a major deterrent to war. Now it appears that this margin is more precarious than had been realized.

It may be argued that the Russians did not invent the jet engine—just copied it. It is true that in 1945 Russia captured many German jets, and later, in the illusory era of good feeling after the war, was allowed to purchase some British Nene jet engines. But the MIG-15 represents a swift advance over these early models, and indicates extraordinary achievement in invention, research, development and production. This goes far beyond mere copying.

Incidentally, the Russians have been extremely canny in Korea. The only important modern weapon they have unveiled has been the MIG-15, and they have cautiously flown it well back of the lines. Consequently the MIG's which we have shot down have crashed in communist territory. Meanwhile

some of our best jets, challenging near the Yalu, have fallen into communist hands. In the ground fighting the Russians have supplied their cannon-fodder comrades—Chinese and Korean—with obsolescent weapons. We, confronted with superior numbers, have had to throw in many of our new weapons, some of which have been captured by the Reds and promptly rushed, no doubt, to the Soviet research laboratories. The Russians have learned more than we in the exchange—another dent in our technical margin.

The Russians are producing atomic bombs. It is true that they stole some of the plans from us, via espionage and treason. But even with this help, as any atomic expert will tell you, they showed a high order of technical skill in getting the bomb into production so quickly. America has an apparently commanding lead in quantity of A-bombs. We should be able to hold it for many years. But how about deliverability? As we have noted, new techniques and inventions could shift the balance there rather quickly—in which case our big atomic stockpile would lose much of its deterrent value.

How can backward Russia compete, inventively and technically, with America, the world's greatest industrial nation? On an all-around basis she can't. But so long as the Soviet Union intensively cultivates the field of military invention, and we neglect it, we are in danger.

Stalin long ago recognized the technical backwardness of Russia and has worked feverishly to repair it through an enormous system of technical schools, research institutes and engineering laboratories. Every year since the 1920's, tens of thousands of the brightest Russian youngsters have been selected for technical training, and evidently the process is beginning to show results.

Stalin also recognized that science, research and engineering are not enough. They can go only so far unless they are fed and fertilized by new concepts, new ideas, new inventions. He reasoned that inventors invent better if they are stimulated by the prospect of large rewards—prizes which glitter in the night, keeping the inventor awake to puzzle over his budding new ideas.

And so from time to time we read in the papers dispatches such as this one of a few weeks ago:

> MOSCOW, Mar. 16 (A.P.)—Another list of 249 Stalin prizes, totaling 17 million rubles, for instrument manufacture, metallurgy and other achievements was announced today.
>
> The list was on top of 141 prizes, totaling 11 million rubles, announced yesterday for scientific research and invention.
>
> The lengthy new list covered five full pages in Moscow's principal newspapers.

Such prizes run as high as 200,000 rubles apiece. However you value the ruble, that is a lot of money to a Russian. Note, too, that the successful inventors get a play in the newspapers comparable to that which we accord movie stars and World Series heroes. The inventor also gets something perhaps more precious to the Russian than money or fame; his "certificate of invention" moves him into the privileged ranks of the technical fraternity, with all that this involves in the way of better housing, food, clothes and service. And the military inventor is the most privileged of all.

Thus Stalin, the realist, in the nationally vital matter of stimulating new inventions, does not hesitate to use the profit motive, with trimmings. He knows that the profit motive, operating through the American patent system, made the United States industrially great. Unfortunately, Uncle Sam, so far-sightedly generous toward commercial invention, has absent-mindedly drifted into a fumbling, near-sighted, penny-wise-pound-foolish policy toward military invention.

America's founding fathers fully understood the crucial importance of invention, and the need for stimulating and rewarding it. At least two of them, Franklin and Jefferson, were themselves men of inventive genius. And so, in the very first Article of the Constitution, Congress was empowered to give inventors "exclusive rights . . . for limited times" to their discoveries.

The founders hated monopoly, but they considered invention so important that they gave the inventor a temporary monopoly—later set by Congress at seventeen years—to enable him to profit from his invention. He could either manufacture and sell the invention himself or he could shop around among competing manufacturers and sell or license his patent to the best bidder. The system is not always equitable and has its abuses, but it holds out glittering rewards which have enormously encouraged American ingenuity and brought to our shores countless valuable foreign inventions to enrich the general welfare.

Naturally it did not occur to the authors of the Constitution, living in the age of musketry and sail, that the survival of this country might someday depend on military invention. They provided no special incentives or protections for the military inventor. Consequently he has been, from the first, under certain inherent disadvantages as compared with the commercial inventor.

In the first place, the inventor of a purely military device has very little bargaining power. His only ultimate customer is the Government. He can take what the Government offers him—or nothing. Or he can peddle his invention to foreign nations, which many Americans have been driven to do. Suppose our Government does not pay him for his invention, but uses it anyway. Until Civil War days, the inventor was not even allowed to sue the Government. Thereafter he had, under certain conditions, a right to sue in the Court of Claims. This used to be an almost interminable procedure. In recent years the court has speeded up its cases, but it still takes a hardy inventor to venture a patent suit there because of (a) the expense, (b) the battery of legal talent and expert testimony which the Government has at its disposal and (c) the difficulty of digging the necessary information out of the Government departments.

Often the military inventor cannot even find out whether the Government is using or infringing his patent. Why can't we bureaucrats get off our pedestals and tell him? Because there is a law (18 USC 198) which makes it a crime for a Government employee to assist anyone in prosecuting a claim against the U. S. A. This law was well intended. It was aimed at a real evil: in the past there were instances where Government employees used their inside information to stir up claims against the Government in order to share in the fees or proceeds. But one curious result is this: even if we in the Government know that an inventor is being cheated of his just patent rights, we can't legally tell him so.

If the invention is so valuable that it is placed under military secrecy, the inventor's position is even tougher. Unless he is specially employed on further development, he can't find out anything. But he can hope. If he lives long enough the secrecy may be lifted. The problem of rewarding inventions on the secrecy list and at the same time protecting military security is admittedly a puzzler. But there are ways in which it could be done.

The military inventor has, besides his inferior bargaining power and shaky legal position, another disadvantage as compared with the commercial inventor. He has—at least in peacetime—a harder selling job. The manufacturer who is shown a new idea smells profits. The peacetime military officer who is shown

a new idea smells trouble. He knows that even the most promising military invention is a long-shot gamble. It will cost a lot of money and many a headache to carry it through the stages of research, development and manufacture. Even then it may prove a flop in practice, thus casting a shadow on the officer's career. The temptation is to play it safe; to give the inventor the brush-off; to say, in the classic phrase of one officer, "Your idea is very interesting, but we are not interested."

An officer assigned to pass on new inventions has a trying job. Inventors are often cantankerous and troublesome. And the pseudo, or crackpot, inventors, who are the most persistent and numerous, are hard to distinguish from the genuine article. After a dozen phony geniuses in succession have twisted your arm and bent your ears, you are apt to forget that the next inventor who comes in the door may have real magic up his rumpled sleeve.

If the invention gets past these first hurdles and surmounts possible roadblocks among the higher echelons, it may eventually come under consideration by the men in the research labs and proving grounds. Here it encounters a new psychological hazard, sometimes called the "N.I.H. factor"—or "Not Invented Here." Since the men in the labs and testing grounds are themselves working on weapon improvement and have ideas of their own on the fire, they are inclined to take a dim view of some outsider who thinks he is smarter than they are.

Some armored-force officers believe this N.I.H. factor kept the Army from making full use of the inventive talent of the late Walter Christie, an American tank pioneer. I cannot pass on this involved controversy. The Army bought a number of Christie's ideas, and it may well be that he was treated with all fairness. But if this inventor had been treated more sympathetically and generously in America, perhaps his models would not have gone to Russia, where they were developed into some of the most formidable tanks in the world. Twenty years ago, I should note on Christie's behalf, distant Russia was not thought of as a military threat to the United States.

Another barrier to military inventions is "No Requirement." Hardheaded, practical, common-sense men say, "We don't need it, couldn't use it." It takes rare imagination to grasp the possibilities of a new invention. Early in World War II someone devised a method for attaching a bulldozer to a tank. It was rejected and forgotten. A tank can knock down or roll over obstacles. Why clutter it with a bulldozer? No one foresaw the hedgerows of Normandy. Within a week after the invasion our tanks were stopped cold by these primitive, massive, root-tangled earthen walls. An earth-cutting tank attachment had to be invented all over again. Precious time and lives were lost, but GI ingenuity saved the day. Sgt. Curtis Culin, Lt. Steve Litton, Capt. James Depew and others improvised a 'dozer-cutter within a few weeks and the tanks sliced forward in the historic St. Lô breakthrough.

Late in World War II, Germany gave us a scare with her snorkel submarine, which was almost immune from radar detection. If she had developed it a bit sooner we would have been in a dangerous fix. Yet this "amazing device" was invented early in this century by Simon Lake, the American submarine genius, and described by him in his U. S. Patent #803,176 for "an air-supply apparatus for submarine vessels." Why didn't we snap up and develop Lake's invention? No requirement. In those preradar days a sub could safely surface at night for air and recharging. Years passed, radar came along, and Lake's ideas continued to sleep in the Patent Office. Then Dutch inventors hit upon the snorkel, the Germans stole or captured it from the Dutch, and soon

the Atlantic teemed with tin fish where no tin fish ought to be.

Now, of course, we have the snorkel. We got it from the Germans. So did the Russians, who also scooped up some of the best German U-boat technicians. The Russians are building a lot of snorkels, and perhaps other new sub types. We believe we can cope with them. We hope so.

In World War II, Germany came up with other "amazing devices" such as the magnetic mine, the acoustic or homing torpedo and jet engines. They gave the Allies a bad time. Yet we had, tucked away in the Patent Office, designs by American inventors for all these devices. (Magnetic mine, 1917, L. J. Husted. Acoustic torpedo, 1926, R. S. Blair. Jets, 1932 and 1934, Urquhart and Goddard.) Read 'em and weep. Each of these inventions would, of course, have required extensive research and development. Somehow, whether from lack of initiative, or money, or imagination, we did not develop them.

Other, poorer nations did. One of these, fortunately, was Britain, from whom we get most of our jet techniques. Britain came up with jet planes in time to counter the swarming German buzz-bombs. The chief British jet inventor was an air-force officer, Frank Whittle, who was specifically assigned to research and invention. Under our present thrifty American system—Executive Order 10,096—Whittle would not have received a penny for his inventions, either from the Government or from the commercial rights, because he was "employed . . . to invent or improve."

The British, however, since their close squeak in World War I, have found that it pays to reward military invention. So the British royal commission, set up to do justice in these matters, cut through all legal technicalities and awarded Air Commodore Whittle $403,000, *tax free*—an independent fortune.

Since 1919, British royal commissions have made awards totaling many millions of dollars to military inventors, individual or corporate. The awards are not based on any narrow view of the patent law—some of the most valuable ideas are not even patentable—but on the contribution made to the national security. Possibly that is why England, with a third of our population and maybe a tenth of our technical personnel and facilities, was ahead of us in many basic military inventions in World War II; and why she was ready in 1940 with the plane types and radar which won the Battle of Britain.

Once World War II got under way, America woke up and performed prodigies of invention, research and development to make up for lost time. Agencies such as the Office of Scientific Research and Development, under Dr. Vannevar Bush, marshaled the nation's scientific and research talent. The stress of war supplied a powerful stimulus for military invention. The armed services suddenly had plenty of money to experiment with new weapons. And Lawrence Langner, of New York, persuaded President Roosevelt to establish the National Inventors Council.

This council was headed by Charles F. (Boss) Kettering, of General Motors, and included many of America's other top inventors. During the war the council and its staff examined 200,000 inventions, and found 6000 of them worth passing along to various Army or Navy bureaus. By the war's end 1000 of these were still under test and 150 had been accepted, including the land-mine detector which saved countless allied lives.

The National Inventors Council still carries on under the chairmanship of Mr. Kettering, and performs a valuable though too-little-publicized function.

Many inventors do not know about it. The military inventor coming to Washington should go to the council's offices, Room 1319 Commerce Building. The council at present has no money to pay inventors or to help them develop pilot models, but it can give them honest advice, steer them to the right channels and save them sweat and shoe leather.

During the war America was also fortunate in being able to apply its accumulated commercial invention—which we have always encouraged—to many military uses. The automotive industry helped out on trucks, jeeps and tanks; commercial aviation—wisely subsidized by air-mail contracts—helped on engines and bombers; the radio industry helped make possible the proximity fuse. The Navy, if a Navy man may say so, did a generally good technical job—partly because we had been allowed to encourage our inventors with some prospect of eventual commercial rewards. Since then we, as well as the Army and the Air Force, have been handcuffed by Executive Order 10,096 and other rules to preserve military inventors against the contamination of monetary reward.

In development and production, America was tops. Britain, for example, was ahead of us in the vital matter of radar. But after we teamed up with the British we were able to improve and mass-produce radar in a manner beyond the capacity of the British, and to the great advantage of both nations.

America's most resounding achievement was the atomic bomb. That was not an invention by an individual. It grew out of the discoveries of many men in many lands. Becquerel and Curie in France. Rutherford in England, and Einstein in Germany started the ball rolling many years ago. From 1939 on, such scientists as Bohr, of Denmark; Fermi, of Italy; Szilard, of Hungary; Chadwick, of England, and German refugees Frisch and Meitner helped encourage America to attempt the bomb, and assisted American scientists in working out the fundamentals. Which should remind us of what we would lose if we "pull back behind the Atlantic." Europe's inventive fertility and scientific genius alone—apart from all questions of honor, friendship, humanity and industrial power—make it essential to have her on our side in the struggle against Soviet power.

Fortunately, America's leaders realized that the atomic bomb was too big to "go through channels." Rules, red tape and false economy were brushed aside. The pressure was on and the sky was the limit. The results were seen at Hiroshima and Nagasaki.

There were other less-spectacular effects. Military men who had formerly looked down their noses at the "long hairs" acquired a tremendous respect for scientists. Military budgets now provide generously for research projects. For the fiscal year beginning this July first, the huge sum of $1,500,000,000 is projected for military research. This is fine, but it overlooks one thing—the inventor. The most elaborate research in the world drifts toward sterility unless it is fed and reinvigorated by new seed ideas, germinal concepts—in other words, by inventive genius.

Invention and science are not synonymous. The quality called inventive genius is sparsely spotted about among the entire population and is rare even among scientists. The popular idea now is that if we spend enough millions on research we can solve any problem we want. There is some truth in this, if we know just what we want. The trick in this is that many great inventions are things which nobody thought of wanting—except that queer bird, the inventor.

Nobody wanted Lake's snorkel. Nobody wanted John L. Bogert's brilliant idea, in 1917, for using carriers to escort convoys, though we got around to it in a desperate hurry a quarter century later. In 1875 electrical science and

research had the techniques needed to transmit speech. The scientists did not think of wanting this. But Alexander Graham Bell did. He made a telephone. The eminent scientist Sir James Maxwell referred contemptuously to Bell as an amateur electrician who had discovered nothing new—merely tinkered together some familiar objects into a device which could talk at a distance. Unfortunately, this disdain of the scientist for the inventor—who is sometimes a mechanic or tinkerer with only a haphazard scientific education—lingers on and colors our thinking today. Actually each is essential to the other. Kettering compares scientists and inventors to the warp and woof of a cloth. "And just try," he suggests, "sleeping in a hammock which is all warp and no woof."

Since Hiroshima the prestige of the scientist has risen sky high. The inventor remains out in the cold. Indeed his position is even weaker, as I say, under the various restrictions, regulations and orders. He also suffers from the idea that invention has now zoomed so far off into the scientific stratosphere that only great teams of researchers can do any good. Therefore, it is argued, the individual or independent inventor is not important any more and the garret or cellar inventor is a joke. This is another half-truth. The germinal idea still arises in the individual mind. And the untutored genius can still sometimes hit the mark where the research teams fail.

Let me give you a few examples from my own recent experience. Since these inventions are within the secrecy zone, I shall have to disguise the circumstances, but I can give enough of the facts to illustrate the point.

One day a small-town auto mechanic came into the Office of Naval Research with a device about as big as your two fists. It is a cheap, simple little thing, made with ordinary tools. But as we examined it and our research people checked it over, we found that this device could do a little trick which nobody had ever thought of before. And this little trick, applied on a scale and scope the inventor never dreamed of, may save hundreds of thousands of American lives in case of global war.

This is one of the rare cases in which the inventor can, I believe, be reasonably compensated. He is not employed by the Government; the technicalities of his patent position are being handled by a good lawyer; and he himself has "actually reduced the invention to practice." This last is most important, because if the Government has to spend a substantial sum of money in reducing the invention to practice, then the Government must—according to regulations—demand a free license, generously leaving to the inventor his commercial rights, if any, when, as and if secrecy, if imposed, is lifted. Get it? Neither do many inventors, who therefore steer clear of military inventions, and devote their talents to such nationally constructive and remunerative fields as dipilatories, pens that will write under water upside down, and bubble gum.

On another recent occasion a veterinarian, without apparent scientific background except among animals, sent us some formulas for making a new propellant fuel. Our trained chemists pointed out that the constituent parts, if mixed as directed, would explode. But the letter was so convincing that they made a cautious try. No explosion. The resulting composition gives every promise of becoming a valuable fuel for missiles.

Here the inventor is out of luck. Since the Government must spend money to "actually reduce the invention to practice," it takes a free license, leaving to the inventor his commercial rights, which are apparently nil. He himself has no facilities for handling a research-and-development contract. Possibly a way can be found for compensating him indirectly, by putting him on

the Government payroll or by some roundabout arrangement with the company which develops the new fuel. It is strange to see Uncle Sam reduced to such shenanigans in dealing with men on whom his survival may depend.

One morning not long ago the mail brought us in a complete set of drawings. They disclosed a weapon of such a revolutionary nature and with such accurate detail and execution that our experts were first startled, then alarmed. Our people had been working along similar lines, and it seemed that this inventor must have got hold of our secret information and then gone on from there. We hastily tracked him down and checked him from A to Z. He turned out to be an obscure Government employee, on a salary of less than $2000 a year, located in a remote part of the United States. He had no special training in science or in abtruse weapons. He had no possible access to secret military information. He just happened to have a strong and original mechanical gift.

His weapon has great possibilities and we are spending a lot of money to develop it. But no money can go to the inventor. In the first place, he is employed by the Government, and admittedly conceived and drew his designs (1) "during working hours," using (2) Government "materials"—pencils and paper. Therefore the Government takes all "right, title and interest" to his invention [see Executive Order 10,096, Jan. 23, 1950, Sec. 1, (a), (1) and (2)]. In the second place, he did not reduce his invention to "actual practice," which he might have done if he had had $1,000,000, private arsenals and proving grounds, and a few other things. But he can still draw his little salary and may someday receive a handsomely engraved certificate of commendation from his grateful Government.

It is true that the Government has an "employee-suggestion" system providing modest—very modest—rewards for bright ideas which save the Government money. Thus R. L. Graumann, an engineer in the Washington Navy Yard, suggested during the war an idea that saved the Government $236,000,000 in the manufacture of 40-mm. shells. He was awarded the munificent sum of $5550. Dick Hoffman, who works for the Corps of Engineers, conceived a pump vent for Army dredges which saved $4,349,000 the first year. He was given $2300. But a new weapon does not ordinarily save money—only lives—so again the military inventor is out of luck.

How did America drift into this dangerous and self-defeating attitude toward military invention, and what can be done about it?

Mainly it is the result of accretion and indirection rather than intention. Our patent system was designed for commercial rather than military invention. Watchdogs of the Treasury, over the decades, created barriers which now protect us better against the ingenuity of inventors than they do against the ingenuity of crooks. And in the long years of our safe isolation behind the oceans, all military expenditures were regarded as wasteful, if not immoral. As for military invention: "A million farmers with shotguns will spring to arms overnight."

During the 1930's and continuing into the 1940's a new factor appeared. The bright young men who swarmed to Washington thought well of Government ownership and were determined to wipe out "monopoly." The eager beavers of the Antitrust Division apparently reasoned as follows: invention means patent, patent means monopoly, monopoly means evil; therefore anything which protects the inventor is evil. They could not see that the patent right is a very special kind of temporary monopoly, designed to benefit rather than injure the public, and for this reason expressly sanc-

tioned by the Constitution. "Patents are bad, Government ownership is good." Therefore the Government should, wherever possible, grab ownership, without compensation, of any patents within reach—and the inventions easiest to reach are those made by Government employees and those having contractual relations with the Government.

In 1943 President Roosevelt asked Attorney General Francis Biddle to make a study of Government policy toward inventions made by employees and contractors. Naturally, Mr. Biddle could not make such a study himself. The task was passed down the line to the bright young lawyers. They worked on it for years. In 1947 they finally came up with a monumental final report of 982 closely printed pages. By then the Attorney General was Tom Clark—now Mr. Justice Clark. He signed the report and forwarded it to President Truman. With all respect to Mr. Clark and the President, I do not believe their busy lives permitted them fully to study, digest and ponder those 982 pages or to understand the harm the report's recommendations might do to the national security. I trust the well-meaning men who wrote the report did not understand this either.

Most of the authors of the report have long since left the Government. But their massive recommendations linger on, carrying the authority of the nation's highest legal officer. They were a primary influence behind Executive Order 10,096, to which I have referred, and exert a continuing pressure against all inventors, military or otherwise, who work for or have dealings with the Government. If only the inventors were hurt, I might just say, "Too bad!" But it is the national defense which is threatened.

So there we are. What can we do?

If we had time to spare, I would suggest that Congress study and overhaul the entire jungle of laws, rules, orders and regulations which are involved. But these are incredibly complex—I have been able only to skim the surface here. Careful revision might take years, and Stalin's boys may not give us that long.

The quickest and simplest move would be for Congress to establish an awards board somewhat along the lines of the British royal commissions which I have mentioned. The British have tested this idea now for thirty-two years and it works. The reports of the royal commissions show that the members, chosen from top technical and scientific men of the nation, have been up against the same kind of sticky problems we face in America, and have solved them. They don't tamper with the existing legal and patent system. But when the existing system provides no reward or an inadequate reward to an inventor, native or foreign, who has well served Great Britain, the commission cuts through the red tape and gives him just compensation.

Since I am not a legislator I would not venture to say just how such an American awards board should be set up. But I suggest that the value of the board would depend on the character and brains of its members rather than on any rigid system of rules. For example, men of the caliber of Conant, Coolidge, Compton, Buckley, Kettering and Waterman could be relied on not to squander public money uselessly. They should be given a competent staff of young technicians, a free hand and a continuing fund of perhaps $5,000,000, subject perhaps to congressional approval of any award over $250,000.

The National Inventors Council has become deeply concerned about our lag in military invention, and at a recent meeting suggested an awards board somewhat similar to that outlined above.

So much for stimulus and rewards. How about better evaluation of new

military inventions? Here I think Congress should give the Secretary of Defense the funds and, if necessary, additional authority to set up special invention evaluation boards in the military services. Evaluation officers should be chosen with the utmost care for their technical brains, their creative imagination and that slant of mind which will enable them to understand that queer bird, the inventor.

If these evaluation boards come upon an invention which seems to have brilliant and useful possibilities, they should not be stopped cold when somebody along the lines says, "No requirement." They should be given the funds and authority to look a little further, to help the inventor build a pilot model and to see that the model gets a fair test.

I have not written this article out of concern for inventors, but out of concern for the United States. The subject is an immensely intricate one, and in trying to simplify it I have made some statements which are open to exceptions, qualifications or argument. But there can be no argument on the main point—that America is not fully utilizing its immense inventive genius in the struggle for survival. I have suggested certain remedies, in the hope that better minds can improve upon them.

SCHEDULING OF WORK IN THE ENGINEERING DEPARTMENT

CAPTAIN R. E. W. HARRISON, USNR

THE AUTHOR

was born in Manchester, England in 1893. Educated at the Manchester College of Technology. Served in World War I in the Royal Engineers and saw service in Egypt, Gallipoli, Sinai, France and Belgium. As an industrial engineer designed and supervised the building of the precision grinding machinery manufacturing plant of the Churchill Machine Tool Company of England, prior to citizenship in the United States. In 1926 Director of Engineering, Cincinnati Grinders, Cincinnati Milling Machine Company. In early '34 appointed Chief of the Machinery Division of the U.S. Department of Commerce. In late 1935 became Vice President, Chambersburg Engineering Company. Transferred in 1935 from Specialist Engineering Branch U.S. Army Ordnance to Lieut. Commander in U.S. Navy as Production Engineering Specialist. Was assigned to active duty as Machine Tool Procurement Officer in the Assistant Secretary's office July 1940 and served in this capacity for two years. Subsequent duties included Special Assistant to the Under-Secretary, operation of the seized Norden Bombsight Plant at Elmira, and the Howarth Marine Thrust Bearing Plant, Philadelphia, operating in both cases as Deputy Officer in Charge for Chiefs, Bureau of Ships and Bureau of Ordnance. Captain HARRISON is a Fellow of the American Society of Mechanical Engineers, and the Royal Society of Arts—also a full member of all other leading engineering societies in U.S.A. and Europe. At present on duty in the Navy Department as Industrial Engineering Officer in the Field Activities Division of the Bureau of Ships.

The engineering scheduling practices discussed in this paper were evolved twenty-five years ago, at a time when the writer undertook to build up a company, making engineering products. At the time when the take-over took place, the company was in a subnormal position, i.e., it had just experienced a year of severe financial loss, its reputation with its customers was not good from the point of view of engineering ability, and in consequence of this, the principal stockholders were in a state of intense anxiety.

Scheduling measures were acutely necessary, because the company's reputation compelled it to accept the less desirable contracts, the characteristics of which were slightly lower prices, shorter deliveries, and in many cases, supremely knotty engineering problems. It is a matter of national record, that the measures adopted by this particular company were successful, and as a result of these measures, the company, within four years, became the leader in its field and the largest supplier of its particular commodity in the world, measured either in units or in aggregate dollars paid by its customers for its output.

This paper will seek to describe only the fundamentals, introduced and rigidly

adhered to, in the engineering department in the carrying out of the company's mission to provide a superior machinery of widely different designs at competitive prices, and equally important, at reliably stated deliveries.

In the build-up from 1926 to 1930, this company's reliability in respect to delivery on time, was as important to the customer, as was the solving of the particular engineering problem, with which that customer was confronted. On the basis that the "proof of the pudding is always manifest in the prolonged eating of same," it is worthy of note that the technique of analysis and the recording of cooperatively arrived at schedules is followed without change today, more than twenty-five years after the system was inaugurated. Furthermore, the concern involved is still the leader in its particular field.

PHILOSOPHY

The underlying philosophy is almost too simple, i.e., the product under review is always the sum of its parts. A reliable concept of the product as a whole, must be based on a study of the parts, or component elements of the program, or product, piece by piece.

After suitable, and above all reliable, identification of the parts, pieces or components, the progressive build-up of these components into a total picture can progress. It should be emphasized at this point, that, just as a modern automobile is the sum of about 3500 major parts, very few of which are strictly alike, so goes the make-up of a locomotive, of a ship, of a domestic dwelling, of a factory, or for that matter, of a Government department.

There is no available shortcut, there is no available royal road. In other words, there is absolutely no alternative to this process of basic analysis, and unless this basic philosophy be accepted, further discussion is futile. Certain eminent engineers, at various points in their careers, have said, "you cannot schedule my brains," and quite frequently when inaugurating an engineering scheduling system, one of the most difficult of the primary problems to be licked is the conviction held by the engineer that his brains cannot be scheduled. This problem can and must be licked and the ways by which this result can be achieved are many and varied.

The most simple of these methods is that of demonstration. Nothing is quite so useful in the handling of a recalcitrant horse as the presence near-by of a well-trained nag. It is common knowledge that, after the wild elephants are corralled, a trained tractable beast is used to lead them out of the corral to a life of hard work, and ordered activity. Successful organizations can provide examples in the way of men and methods, demonstrating that a scheduling of engineering is a practicable operation, and it is suggested that opposition to the idea where encountered, be caused to melt away by demonstrations of success in the matter under dispute.

It is a *sine qua non* that all the cooperation in the world frequently cancels out to a big embarrassing zero, unless there is a directing brain coordinating the cooperation; in other words, cooperation without coordination is just a demonstration of a mass of amorphous goodwill, which is about as effective in demolishing an engineering problem as would be a custard pie shot out of a gun in an endeavor to do the work of a projectile. So, we reach the point—what is scheduling? It is nothing but intelligent coordination. How do we arrive at it? First, by detailed analysis. Secondly, by the building up of the successful steps of fabrication, with all steps coordinated so that they overlap and thus conserve time wherever possible, but above all, that they follow

each other in logical sequence, with all available guns trained on the one ultimate target, i.e., completion of the project on or before an agreed-on date.

The principal enemies, which have to be overcome before this philosophy can be put into effect, are, sad to relate, present in about the following order:

1. Prejudice.
2. Ignorance, stemming from lack of appreciation of the value of adequate analysis and scheduling.
3. A *laissez faire* attitude, springing from indifference.
4. The failure of the head man, or boss, to realize that when things go wrong, or programs fail to materialize on time, the fault is fundamentally his, in that he has failed to supply the leadership, which should, and could, correct the deficiency.

At the present time, with practically every major defense program either behind time, or being rescheduled for a new and later delivery date, it is high time that those who are making a contribution, or who have a responsibility, take another good, long, hard look at what we have before us, bearing in mind that our potential enemy in any forthcoming conflict, now has the benefit of Germanic thoroughness of analysis, coupled with the drive which people invariably have after twice suffering defeat, and determined that if they alone cannot achieve the ultimate defeat of the victor, they will at least participate in the ultimate victory, if only as subordinates on the side of the victor. Sometimes this author thinks that we underrate the terrific head of steam which turns the wheels of the forces which we could have to defeat, and while detailed exposition of the factors in this situation would be out of place in this paper, the point is mentioned, reiterated, and emphasized, in support of the plea that we put our own house in order, and run interference for that most valuable asset which we possess, i.e., our Yankee ingenuity for the production of good and valuable things on a large scale and at an economically acceptable price.

The production wonders in the way of most volume, good quality, and accurate schedule achieved in the last war, were definitely not the results of happenstance. They were definitely the products of the brains of hardworking, dedicated people, motivated, first, by patriotism, closely followed by the realization that never again would they have the opportunity to exercise their unique production genius, if the U.S. were to be defeated in the then raging battle.

The judgment of those best able to appraise the value of the factors in the current situation is that the threat today is infinitely greater than it was when we faced the Germans and the Japanese. In the forthcoming conflict we cannot look forward to a superiority of numbers, such as those that we ultimately achieved in World War II by the attrition of the enemy, and we must, therefore, achieve our ultimate victory by using only *good management.*

The factors which go to make up good management are legion. I propose in this paper to discuss only good scheduling as a component of good management in the engineering department.

It is astonishing to me to find, in the oddest places, a lack of respect for the value of good scheduling, and the oddest place of all to find this lack of respect for good scheduling is in a military establishment. Why? A good production schedule is nothing more or less than an operational order, and God knows, officers of all Services spend their peacetime lives going to school with no other basic objective in mind than the creation of an ability to write a good operational order.

* * * *

The writer believes that the foregoing is sufficient description of the "character of acceptance" which must be generated before attempt is made to achieve a good schedule; however, before getting down to the brass tacks of actually doing the job, a word is desirable for the benefit of those who can see only the difficulties, which they believe in the aggregate are insurmountable. To systematically demolish the obstructions, it is just as necessary for the job to be tackled systematically, sincerely, and without let-up, as it is for the production schedule to be built up by benefit of the same virtues and practices. The two situations complement each other. To build up a good step-by-step production schedule is, after all, only half the battle. Concurrently, the offensive must be waged relentlessly on the opposing forces represented by such "thorns in the side" as contractors' delinquencies, subcontractors' delinquencies, inspectors' failure to recognize the key nature of that element in the program entrusted to their care, and above all, the failures of the leaders to whip their crews into a state of aggressiveness until they reach the point where they simply refuse to be denied. By all measures short of illegality, or poor ethics, the detracting factors must be demolished and kept demolished. *Nothing can be assumed to be automatic in this process.*

It is possible in normal business life, in the manufacture of a mass production product, such as a telephone or a motor car, or an electric motor or a refrigerator, all made by the hundreds of thousands on the production lines, for an IBM system to be devised, which will throw up a red ticket as a danger signal that the production schedule is about to go out of whack. However, there are very few of the defense production problems that are being wrestled with today where reliance can be placed on such a delightful panacea as reliance on a red ticket popping out of an otherwise inarticulate machine. To assume that the machine can take the place of the foresight, drive and perception required of the sentinels on the scheduling production line is pure fallacy. There is no substitute for hard, conscientious work on the part of those responsible for the *completion of the program.*

There are all kinds of discipline. We like to think that we are intellectually immune to the physical discipline which corrected the views of those of us who, in their youth, transgressed the laws of the home, but the fact remains that unless we accept and practice intellectual discipline, whether it be competitively commercial or Government departmental, we will, as sure as God made little apples, eventually suffer an infinitely more painfully applied physical discipline, perhaps in the form of the labor camp, the concentration camp, and all the futile brutalities that our alleged liberator from Capitalism would thrust upon us.

Proper scheduling of engineering work is synonymus with good planning, and in these days when a material allocation under rationing is often only a hunting license, carefully detailed scheduling in the initial stages, and a rigid adherence to such scheduling are a prerequisite to meeting or anticipating delivery dates.

* * * *

The following are some examples of basically good and successful scheduling:

1. Building a factory.
2. Building a locomotive.
3. Building a machine tool.
4. Building a ship.

It may seem strange to those knowing the writer's current interest and responsibility that the problem of building a ship is cited last. There is, however, a method to this madness. First, this paper must build a case, and it is hoped that that result has been achieved, i.e., the case for engineering scheduling in detail.

BUILDING CONSTRUCTION

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	TEMPERATURE																																																														
	WEATHER																																																														

The integrity of this building erection schedule can only be preserved by the availability of working drawings and specifications ahead of the specified times when each segment of the program is due to start. Overall planning of the building operation is predicated on this assumption.

Thereafter, management of the project resolves itself into management by the rule of exception, i.e., the manager devotes his time and talents to dealing with unusual and unforeseen difficulties, which, if not dealt with promptly, would hazard the completion date.

The primary scheduling stages are:

a. General specifications
b. General arrangement drawings
c. Detailed specifications
d. Detailed drawings as necessary.

Failure to schedule completion dates for each of these preliminary stages, and to rigidly adhere to such schedules can, and frequently does, delay production of necessary bits and pieces. The result is that scheduled assembly or construction may proceed smoothly up to a point. Then nonavailability of a part halts the operation as well as other segments of the work. Men are released and later must be reassembled and recalled to complete the job.

The inevitable result is that at the end of the construction or assembly period, a congestion of work occurs. Jobs interfere with each other, and inefficiency results. A manpower peak, accompanied by overtime, aggravates the condition and increased cost as well as delay ensues.

Let us next take a look at these schedules which have proven successful for building such widely diversified products as a locomotive, a factory, a machine tool, or a ship. What are the common factors, and wherein in these various schedules is the foundation laid for the production of a usable result with performance in accordance with design specification and completion with an agreed-on product delivery date? What are the common factors? In the author's humble opinion these factors, or steps, are involved:

Step 1: Establish a date by which the final result to be attained is to be agreed upon and completely spelled out. If, as frequently happens, the factors which go to make up this desired result can only be determined after experiment and laboratory tests, then, the first task is the scheduling of the writing of the performance specification of the finished product. Sometimes the collective experience of the people engaged in the project can be drawn on to such an extent that it is possible to write a performance specification which is achievable without the aid of new laboratory research. All well and good, if such is the case, but if the reverse be true, then the first segment of the problem of attack must be the scheduling of the steps to be taken to achieve the writing of an achievable performance specification. This is common ground to all engineering design problems.

Usually an exploration will determine the existence of a key element. In the case of the machine tool, it is usually power, plus the ability of a cutting tool capable of sustained delivery of that power. In the case of a diesel electric locomotive, it is the ability of an engine to convert the BTUs of the fuel into tractive force exerted on the rails. In the case of a factory, it is frequently the reliable availability of the labor to operate that factory. In the case for a ship, it is the combined performance of the hull in the water, and the ability of the propulsion machinery to propel the ship at the desired speed in all sailable waters.

Step 2: Establish a date by which the following must be completed.

a. General arrangement drawings
b. General specifications
c. Detailed specifications
d. Necessary detailed drawings.

Step 3: Establish a deadline date for completion of the bills of material. This requires itemized consideration of that which is needed in the way of bricks and mortar, of iron and steel, of bronze and aluminum; in fact, each and every ingredient needed to produce a product which will make available a performance in accordance with (a).

* * * *

These are not normal times. In normal times the creation of a new product can be pretty much of a foregone conclusion, if the engineers and plant managers know their basic business of design and management, but today, and in the coming days which are likely to

MACHINE TOOL DESIGN SCHEDULE

TASK	CALENDAR																																				
	January																			February																		
	F	S	M	T	W	T	F	S	M	T	W	T	F	S	M	T	W	T	F	S	M	T	W	T	F	S	M	T	W	T	F	S	M	T	W	T	F
	17	18	20	21	22	23	24	25	27	28	29	30	31	1	3	4	5	6	7	8	10	11	12	13	14	15	17										
1. Prepare specification of required performance																																					
2. Make work capacity diagram																																					
3. Make power requirements diagram																																					
4. Make coolant and overall lubrication diagram																																					
5. Draw up machine outline along exploratory lines (full size)																																					
6. Make exploratory unit designs																																					
7. Make integrated machine design with blocked in functional specifications for each major unit																																					
8. At this point check ability of design to meet performance specification																																					
9. Prepare first unit design																																					
10. Detail same																																					
11. Prepare second unit design																																					
12. Detail same																																					
13. Repeat for all other major units																																					
14. Check for ability to assemble																																					
15. Check for functional interferences																																					
16. Write in specifications for tolerances and finishes																																					
17. Write in material specifications																																					
18. Check for completeness of all drawings and specifications																																					
19. Run appropriate laboratory tests on units or components where necessary																																					
20. Report completion of assignment to appropriate authority																																					

NOTE: The calendar attached to the assignment naturally varies in length according to the size of the task undertaken. Where, as is frequently the case, contractual delivery requirements call for a minimum required time of delivery, managerial skill in the design department must be used in arranging for overlap of drawing production operations in such a way that there is a steady flow of required basic information. In almost all cases, a minimum 10% time cushion is a prerequisite to provide flexibility in the management of the design schedule.

prevail for the next few years, something more than this is required, i.e., we need an ability to fit the operational management of the enterprise into the pattern established for us by a rigid Government control of nearly every element which goes into the ultimate picture. A manufacturer today, who is a contractor, who has obligated himself to supply the Government with ships, must fit his pattern of management into a rigidly delineated picture which:

a. Writes the specification for performance of the end product.

b. Controls all the major phases or conditions of employment of his labor, such as hours of work, maximum salary, minimum wages, selection of materials, rationed procurement of materials, and in some cases, the very location of his

plant. What is left, you may well ask, wherein the manufacturer can exercise his talent for management?

The answer is *"plenty."* It is a matter of prime necessity in these days that no matter what political party be favored, the fact be recognized that we operate very largely under Government control, and to operate successfully, the company's procedures must be fitted and fastened into this fabric of national pattern; otherwise, conflicts of interest and failures to coordinate will characterize every phase of the production operation. No matter how much an individual may resent the existence and imposition and intrusion of these controls, which greatly reduce his personal liberty of thought and action, it is obvious that the first struggle he has got to have, is with himself, in that he must convince himself that he is powerless to change the overall picture, and the sensible thing to do is to put the few advantages to work in his favor, and to take steps to see that the numerous disadvantages in this multitudinous mass of controls, embarrass him and endanger his schedule to the least possible extent. Failure to do this, results in individual ulcers and broken contracts, but more importantly, it reduces the country's ability to defend itself, and that, is a matter of prime seriousness for everyone.

* * * *

There are just as many equally applicable do's and don'ts which must be observed by the procurement or contract generating agency, if required goods and hardware are to be secured on time. In other words, the requirements which can and should be specified by the originating agency can be just as well defined, whether that agency be governmental, or a unit of private industry. For instance, adhering strictly to the subject of the scheduling of engineering services, it is inevitable that at all major and intermediate stages in the production of a design, important and correlated decisions must be made as promptly as circumstances permit. Without these clear-cut decisions, the contracting agency cannot function.

In private industry little difficulty is encountered in this phase of processing engineering work, inasmuch as if a roadblock develops, its existence quickly becomes apparent, and corrective measures are usually obvious. In some other activities corrective measures are not quite so easily generated, although they are just as necessary, and it is a part of the duty of the supervisor or chief of unit to see that they are put into effect.

Where the situation is further complicated by the presence of a military type overall supervision, a prolific source of delays sometimes develops due to the necessary system of rotation of officers through a wide variety of duties. Usually the tours of duty are relatively so short that close on 50-percent of the time is taken in learning enough about the job to be able to operate.

Under the circumstances continuity can only be achieved, obviously, through the civilian staff, hence, a major contribution can be made by building up its professional quality, compensating at a rate which makes the positions highly desirable, but above all, creating an atmosphere of professional alertness and competence which will automatically accept a schedule, once agreed upon, as the bible and yardstick for the whole operation. There is always plenty of room—at the top—in any organization for a contributor of brains, and the military in this case, is the natural pipeline or channel through which should, and will, flow the basic ideas and seagoing experiences, which are of the very essence of the design mission.

There is an inevitable tendency, in all organizations, to demand a record of precedent in support of any major move, but there also inevitably comes a time when this corroborating evidence just does not exist—action is imperative, and someone, i.e., the natural leader,

SHIP DESIGN

PLAN SCHEDULE 440

Item	No.	Man Days		1951							1952			
		Est.	Act.	J	J	A	S	O	N	D	J	F	M	
CODE 441	CVB No. 59 S0101													
Gen. Arr. 1.B, H, 2nd Plat.	601743	180		←	—	—	→							
Gen Arr. 1 Ft. Pl. 4,3 Dk.	601744	200		←	—	—	—	→						
Gen. Arr. 2nd, Main Deck	601745	200				←	—	—	—	→				
Gen. Arr. 01, 02 Levels	601746	180				←	—	—	—	→				
Gen. Arr. Gal., Flt. Deck	601747	200						←	—	—	—	→		
Gen. Arr. Inbd. Pro. Sect's.	601748	150						←	—	—	→			
Gen Arr. Outbd. Pro. P/S	601749	150								←	—	→		
Miscellaneous Studies		400		←	—	—	—	—	—	—	—	→		
Amm'n. Stowage, Ship	601750	150					←	—	—	—	→			
Amm'n. Stowage, Air	601751	150		←	—	—	—	—	→					
Amm'n. Handling	601752	150		←	—	—	—	—	—	→				
Amm'n. Misc. Studies		270		←	—	—	—	—	—	—	→			

Scheduling the plans for a carrier entails an enormous amount of detailed design and drafting, which must be integrated into an overall picture which culminates in a reliable delivery date. There is no adequate substitution for sound experience in the preparation of such a plan schedule By recording the actual man-days consumed as against the estimated, an experienced observer can forecast man-day expenditures and total time requirements to a remarkable degree of accuracy.

The man-day expenditures for individual assignments become, in course of time, relatively easy to estimate; the key requirement, however, is integration and the will and necessary authority to speed up the lagging elements, thereby preserving integration and assuring completion of the whole on or before the deadline date.

Acceptance of this principle is the only basis for integrity of the system

must move out and take the responsibility—to dodge it counts disaster.

At this point, herein comes the laboratory and the test plant, and the history of progressive engineering is replete with examples of courageous action, stemming from which we have modern welding, high pressure, high temperature steam, the laminating of wood, and the shaving of propulsion gear teeth, to quote but a very few.

Apart from the problem of securing supervisory personnel with the courage to act promptly and with good judgment in respect to new materials, practices and procedures, there exists the collateral problem of staffing the engineering organization with section and division

leaders who will regard an engineering design production schedule as something which must be lived up to, or alternatively died for in the effort to maintain it.

In seeking out, appointing, guiding and inspiring such a staff, the personal attitude of the Branch Chief is the key factor of paramount importance. Unless this official is himself fanatically sound on the subject of coordinated effort as expressed in the form of a well thought out schedule, the whole scheme is doomed to failure before it gets underway.

As noted earlier in this paper, the preparation of engineering drawings and paralleling specifications in accordance with an agreed-on time schedule is the very foundation, and the only foundation, on which an engineering performance can be predicated. Any failure of any link in the chain of principal events is multiplied in effect 1000 fold in the work shops, the works offices and on the building ways. In fact, it might be said that a chain reaction sets in from any delinquency at engineering headquarters, this reaction mushrooming into a cloud of disorganization which, in turn, sets back and disorganizes the work of the whole of the production team, ranging from engineering concept down to final acceptance test.

Material procurement is not only a major problem today, but it is likely to remain as one for a number of years to come. Obviously material requirements cannot be developed on schedule if the engineering work on which they are predicated is out of phase. No amount of follow-up or expediting action will ever regain those lost days of time.

In those cases where a structure must, of necessity, be put together from locally made components coupled with machinery components produced by outside contractors, it is just as important for the outside contractors to adhere to the master schedule of principal events as it is for the shops and offices within the parent organization. The availability of funds to finance outside purchases is seldom a problem in private industry, but in Government establishments, where funds are made available on a fiscal year basis, considerable forethought and courageous estimation of the situation must be exercised so that delivery schedules for goods bought on the outside synchronize with those planned for the home shops. At this point it is pertinent to say that it is just as necessary for consideration to be shown to contracting manufacturers by giving them as much advance information as possi-

SHIP ERECTION

PRINCIPAL EVENTS

Award	**61**
First Structural Plans to Yard	**93**
First Steel Received Section No. 1	**105**
Start Subassembly	**107**
Start Installation of Piping in S.A.	**110**
Lay Keel	**113**
Start Installation of Seachest	**120**
Skeg Castings to AH	**128**
Stem Casting to AH	**135**
Start Tank Testing	**138**
Start Installation of Auxiliaries	**138**
I.B. FWD Struts to AH	**138**
I.B. Aft Struts to AH	**142**
Run Preliminary Line	**146**
Start Installation Gen. and Cond.	**149**
Start Erection of Boilers	**150**
Start Installation of Main Mach.	**150**
Hawse Pipes to AH	**165**
Install Rudder	**180**
Run Final Line	**183**
Launch	**200**

The calendar of principal events sets the tempo for the Engineering Department. Lack of approved plans, at the time when they are needed, automatically makes a hash of the overall planning. Sheet anchor or bible—call it what one may—all Departments and primarily the Engineering Department, must anticipate this calendar.

ble, as it is for this same type of information to be given to one's own shops.

The delusion still persists in many quarters to the effect that a supply system can function automatically if well organized, but manned by non-technically informed help. The contrary is the case. While IBM machines can do a lot of marvelous tabulating and integration of tabulated data, the machines definitely cannot think, and when thinking has to be done, it should be of high caliber. It is just as true today as it ever was in Government and industry, that more and better work can be accomplished by few well-trained thinking individuals, than can be accomplished by a multitude of people neither trained nor capable of thinking.

In conclusion, two wrongs can never make a right, and no amount of expediting can compensate for the time lost by getting a contract off on the wrong foot, due to the fact that the contract could not be defined, because the engineers had failed to deliver their quota. Conversely, the presence of a lot of expeditors on a given job is automatically an admission of poor scheduling out of which stems ineffective planning.

The pattern of successful engineering management is unmistakable. The principles on which this success is founded are immutable.

Not even the great automobile industry of the United States has witnessed so phenomenal a growth and such markedly successful engineering achievement as has the business of building diesel electric locomotives.

The largest and most successful diesel engine locomotive building company in the world describes the scheduling of its engineering design work as follows:

"The Engineering Department meets with Management, Sales Department and Manufacturing Division regularly each month and at the beginning of each year for the purpose of laying out a program for those items which are to be accomplished by the Engineering Department during the ensuing year. Each of these items discussed are assigned dates for completion of the project. In the following meetings the progress made towards those completion dates is continually followed up. This gives the general overall engineering program.

"However, this may not apply only to a particular locomotive design, but can also apply to any component or development thereof. The design of a new locomotive model is roughly blocked out in meetings with Management and the Sales Department. Consideration is given to the customers' requirements or demands and the best way of supplying these are determined, providing a suitable market exists. Further meetings with Sales and the Manufacturing Division are then held, to set down the lead times for the various components of the locomotives. Working back from the established dates of completion, the lead times determined will give a guide as to what component drawings need be completed first. A complete engineering schedule is then laid out by drawing description, together with the completion dates established for each item. This is further extended to become a complete list of drawings by part numbers as the scope of each drawing is determined. The completion of this drawing list is then followed by weekly meetings within the Engineering Department itself until the final drawings have been completed. Each drawing is routed and scheduled on to the drafting board, off the board through checkers and corrections, into records, and then out of the Engineering Department as a final release to those departments requiring them.

"We have actually found it feasible to predict the actual number of engineering man hours required to design a new model and have held to this schedule, have been able to time it and cost it so as to meet estimated completion dates and budget requirements."

SHIP CONSTRUCTION

Section	Plans		Material		Molds		Layoff		Fabrication	
	Start	Finish	Start	Finish	Start	Finish	Start	Finish	Start	Finish
1	93	96	101	110	94	112	102	117	103	120
2	97	125	105	122	98	124	106	129	107	132
3	99	105	107	124	100	126	108	131	109	133
4	100	107	108	125	101	127	109	132	110	135
5	101	105	109	119	102	121	110	126	111	129
6	102	106	110	121	103	123	111	128	112	131
7	104	108	112	127	105	129	113	134	114	137
8	105	110	113	129	106	131	114	136	115	139
9	106	111	114	132	107	134	115	139	116	142
10	107	114	115	134	108	136	116	141	117	144
11	110	115	118	133	111	135	119	140	120	143
12	111	114	119	127	112	129	120	134	121	137
13	112	116	120	135	113	137	121	142	122	145

Section	Sub-Assembly		Erection		Fitters	Drillers	Riveters	Welders	Search	Test
	Start	Finish	Start	Finish	Comp.	Comp.	Comp.	Comp.	Comp.	Comp.
1	107	119	113	123	127	129	131	133	138	142
2	111	131	117	135	140	142	144	146	148	150
3	113	132	119	136	141	143	145	147	148	152
4	114	134	120	138	142	144	146	148	150	154
5	115	128	121	132	136	138	140	142	144	146
6	116	130	122	134	140	142	144	146	148	150
7	118	136	124	140	146	148	149	151	153	155
8	119	138	125	142	148	152	154	156	158	160
9	120	141	126	145	151	153	154	156	158	162
10	121	143	127	147	153	155	157	159	161	163
11	124	142	130	146	154	154	156	158	160	164
12	125	136	131	140	146	148	150	152	154	156
13	126	144	132	148	153	154	155	157	159	161

NOTE: The approved production engineering scheme starts with the provision of the plans and actual production. The integration of concurrent operations is particularly noteworthy. Failure to provide approved plans on time destroys this carefully thoughtout integration at the inception of the shipbuilding operation.

The injection of plans covering alterations to the original concept at any time during the building period also automatically disintegrates the production schedule from the date of receipt of the revised plans.

Successful shop planning of operations can *only* be achieved when the drawing office, or engineering department, production schedule, covering plans and specifications, can be implicitly relied upon.

CONCLUSIONS

Machinery, as such, is easy. It is people who are difficult. The well-trained engineer who knows his materials and processes can produce his designs and specifications with almost clock-like regularity. If care is exercised in the selection and maintenance of the engineering force, the quality of the design and specification work is assured.

The timing of the production of this work, unit by unit, is one of the highest phases of engineering management of men, and it is in the failure to recognize this factor from whence many of the present day difficulties stem. Very frequently the cause is self-evident, i.e., the promotion system which, through the workings of seniority, takes a good engineer off the drawing board and promotes him into a management or scheduling position, for which he is neither trained by practice, nor equipped by instinct. It is believed by the author that if this one factor alone be rooted out and corrected, 90 percent of the problem will be solved.

It is a fair estimate that the remaining 10 percent sits on the doorstep of the Chief Engineer, or his counterpart. If schedules are as important in a key way, as in our hearts we know them to be, then the Chief must manifest his interest in them visibly. It is futile and inadequate for him to say—"my subordinates take care of this matter." They just do not. If the schedule is to be the bible, it must be laid down in bar chart, or other graphical form, on the walls of the office of the directing head. It must be clear-cut, simple and up-to-date. In the automobile industry such charts are maintained by the hour—in most other private industries, by the day.

All engineering production can and should run with the precision of the timetable of a first class railroad, and it is a *sine quo non* that the same quality of brain power which today devises our metropolitan railroad time tables can bring to bear the same thoughtful organizational thought on the engineering work-scheduling for the production of which they are responsible.

Profit earning history in industry in those cases where the history is one of consistent success, is at all times irrevocably tied in with the principle of management by the rule of exception, i.e., the big boss, the big wheel, presumably the best brain, only gets in on the routine daily stint when something goes wrong, or something new must be added or something old subtracted. Generally speaking, these men are never too busy to do the detached, constructive thinking which is a must for the head of a progressive organization. Quite frequently the heads of our largest corporations operate these corporations, insofar as the bulk of their current business is concerned, on the strength of not more than one-half dozen well thought-out and utterly accurate daily reports.

Trouble signals are automatic and unmistakable. Corrective action follows promptly. A $100 mistake in automobile design on automobiles coming off the production line at the rate of 1000 a day, means automatically a loss of $100,000 a day, or one-half million dollars a week. When the economic structure of an industrial organization starts to bleed at this rate, anemia and death follow fast. When the economic structure of the body politic of a great republic bleeds at the same proportionally high rate, one can almost hear the carrion crows on the fence sharpening their claws and the beasts of the forests whining hungrily in anticipation of the feast which will be theirs.

THE DISTINGUISHING CHARACTERISTICS OF BOILER TUBE FAILURES

FRANK E. CLARKE

THE AUTHOR

graduated from Western Maryland College in 1935 with A.B. degrees in chemistry and education. Studied advanced physical chemistry for one year at University of Maryland and taught chemistry for four years before entering the Boiler Water Section at the E.E.S. Chemical Laboratory in March 1941. He became Head of the Boiler Water Section in 1942, Head of the Chemical and Physical Branch in 1946 and Head of the Process Branch of the Chemical Engineering Laboratory in March 1951. Since 1942 he has been responsible for Naval boiler water treatment and diagnosis of failed boiler tubes submitted by the Fleet. He is one of the Navy's principal consultants on water treatment and boiler problems.

Diagnosis of boiler tube failures is an important duty of ship's personnel and boiler repairmen as well as of the technical laboratories which deal in this service. Accurate interpretation of tube damage at the time of failure or overhaul may disclose sources of damage which the laboratories only can surmise, and thereby provide valuable background for recommendation of corrective measures. Boiler operators and maintenance men can make very acceptable appraisals of tube failures by examining the damaged areas for pertinent characteristics. This paper discusses the distinguishing characteristics of the principal types of failures. It is aimed at promoting better understanding of boiler damage in general and more accurate and informative reports on boiler casualties.

CAVITIES AND SCARS

This category includes all pits, grooves, gouges and wasted areas which have resulted in obvious internal or external loss of tube metal, without evidence of abnormal temperatures. It covers localized corrosion, general corrosion, steam gouging and mechanical scars.

Dissolved oxygen pitting is the most localized form of cavitation. The pits are straight walled or conical, depending primarily on the rate of attack, and range from 1/16" to 1" in diameter. Active pits are filled with compact, hemispherical scabs of corrosion products. These are generally reddish-brown on

FIGURE 1.—Oxygen Scab, 2" Generating Tube.

FIGURE 2.—Oxygen Pit, 2" Generating Tube.

FIGURE 3.—Oxygen Pit, 1" Superheater Tube.

FIGURE 4.—Oxygen Pits, 1" Desuperheater Tube.

their water sides, but are black with magnetic iron oxide on their interiors. Inactive pits have the same general shape as active pits except that they have no scabs and are reddish-brown in color instead of black. Clean inactive pits indicate that earlier dissolved oxygen corrosion has been arrested. Inactive pits in themselves have very little tendency to promote further corrosion.

Dissolved oxygen pitting causes about 30% of all Naval boiler tube failures. The most serious examples are found in the water wall tubes, fire row tubes, desuperheater tubes and superheater tubes of boilers equipped with open or semi-closed feedwater systems. Figures 1 to 4 show examples of typical pits. Similar pitting will occur in generating and desuperheater tubes of boilers equipped with efficient deaerators if the deaerators are operated improperly, if the boilers are fed with aerated reserve water instead of deaerated feedwater, or if they are secured at steaming level so that air leaks in on cooling. Pitting in a superheater means either that it was filled with aerated water on securing the boiler, or that it was secured without proper draining so that aerated condensate puddles formed when the steam condensed. Puddle pits generally can be identified by the fact that the holes are surrounded with stained areas representing the shore lines of the puddles. Such a shore line can be seen in Figure 3.

Fire side cratering, like that shown in Figure 5, resembles dissolved oxygen pitting in that it consists of deep straight walled cavities. These scars differ from oxygen pits in that they are irregular holes and meandering, worm-like tracks rather than cylindrical or conical holes. Fire side cratering is caused by water trickling down the tube exteriors or standing around the tube ends. Oxygen from the furnace atmosphere and acidic compounds from fire side deposits intensify the effect of the water. This type of damage occurs most frequently on the lower ends of water wall and divi-

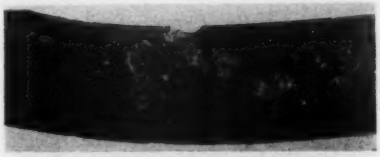

FIGURE 5.—Fireside Cratering, 1" Wall Tube.

sion wall tubes, where plastic refractory has shrunk and created water traps. It has been a recurrent problem in the water walls of inactivated "D" type boilers, apparently because moisture condenses from the furnace atmospheres and collects at the tube ends.

FIGURE 6.—Steam Gouge, 1½" Generating Tube.

Steam gouging is a more remote type of localized cavitation. It occurs when steam from a perforated tube impinges on an adjacent tube and erodes and corrodes the metal away. Unlike oxygen pits and fire side craters, steam gouges have smooth, oblique walls and soft passive surfaces. A typical steam gouge is shown in Figure 6.

Waterside grooving is a semi-localized type of cavitation. It occurs primarily at the water drum bends of fire row generating tubes. It consists of deep longitudinal cavities, several inches long and up to one inch in width. The cavities sometimes are filled with flaky scabs of black iron oxide. Heavy accumulations of sludge can cause grooving by partially protecting the tube metal from the boiler water and thereby producing electrolytic concentration cells, where the protected area acts as the anode and the remainder of the tube acts as the cathode. Dissolved oxygen increases the effect of concentration cells. Film boiling in hot spots of fire row tubes can produce the same damage by concentrating caustic in those areas, and thereby dissolving the protective oxide film as rapidly as it forms. The source of water side grooving generally can not be determined by visual inspection of the damage. A typical example of grooving in the water drum bend of a generating tube is shown in Figure 7.

FIGURE 7.—Waterside Groove, 1½" Generating Tube.

FIGURE 8.—Fireside Groove, 1" Superheater Tube.

Fireside grooves also are a semilocalized form of cavitation. They are most apt to occur near the header junctions of horizontal superheater tubes and cross drum boiler tubes. They consist of narrow circumferential grooves from ½" to 1" in width, as shown in Figure 8 of a superheater tube end. Damage of this type almost always results from a leaky tube above dripping water on the affected tube. The deepest part of the groove is directly under the leak. External grooves of a very similar type can form at the water drum ends of vertical generating tubes, due to the accumulation of shallow soot deposits. These soot grooves differ from water grooves in that they affect the entire circumference of the tube uniformly and usually extend over a greater tube length.

General waterside corrosion is manifested as a relatively uniform loss of tube metal over a considerable length of the tube. The damaged area is covered with innumerable, tiny, overlapping pits, as though the metal had been sandblasted. Wall thinning generally is greater on the side nearest the flame. Low alkalinity is the most common cause of this corrosion. It is rare in Naval boilers. Figure 9 shows an example of it in a condensate line. Similar corrosion has been found in fire row, generating tubes.

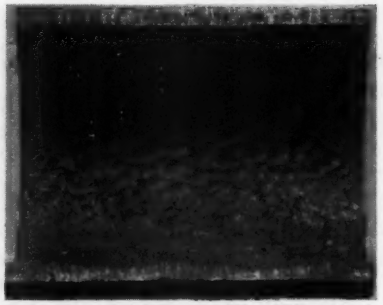

FIGURE 9.—General Corrosion, 3" Condensate Line.

Soot corrosion is the most generalized type of cavitation on the fire sides of boilers. It gets its name from the fact that it occurs primarily at the water drum ends of generating tubes and the header ends of economizer tubes, where moist, acidic soot deposits imbed the tubes. Soot corrosion is characterized by more or less uniform circumferential thinning of the tube exteriors for distances ranging from several inches to several feet. The affected metal generally has a greyish-brown, passive, sandblasted appearance. The severity of soot corrosion is influenced by the amount of moisture which the soot absorbs from the furnace atmosphere or tube leaks. A typical example of soot corrosion is shown in Figure 10. This type of damage causes about 20% of Naval boiler tube failures.

FIGURE 10.—Soot Corrosion, 1" Generating Tube.

TUBE DEFORMATIONS

Deformations are distortions of the tube metal caused by plastic flow, thermal stress or mechanical stress. They do not involve loss of metal as in the case of cavitation. Heat blisters, thin-lipped bursts, thick-lipped bursts, sagging and warping are the most common tube deformations. Together they account for more than 30% of all Naval boiler casualties.

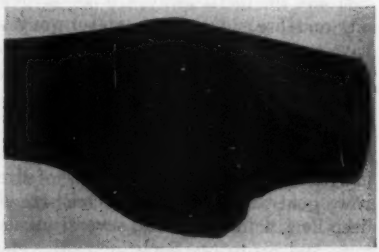

FIGURE 11.—Thin Lip Burst, 2" Generating Tube.

Thin-lipped bursts of the type shown in Figure 11 are the most spectacular form of tube deformation. These are large gaping holes which resemble bursted bubbles. Their open lips are uniformly tapered to sharp knife-like edges, with no evidence of cracking or tearing. Deformation of this type means that the tube metal was heated suddenly to softening temperature so that it melted, blew into a bubble and burst

under boiler pressure. True thin-lipped bursts occur in economizer tubes, generating tubes and to a considerably lesser extent in alloyed superheater tubes. In every case they indicate that the water or steam flowing through the tube was far below the quantity necessary to absorb the heat poured against it. Tube stoppage by foreign objects, such as tube brushes, gloves and tube plugs, is the most common cause of this type of failure. Some authorities consider that stoppage is the only source of thin-lipped bursts in generating tubes; however, there is considerable evidence that simultaneous increase in steam demand and firing rate or momentary low water can cause them. Soot fires after securing the furnace are an occasional cause of thin-lipped bursts in economizer tubes. The fire generates a localized, intense heat which turns the economizer water to steam and melts the tube after the feedwater flow has stopped. In a superheater, such bursts can result from reversal of superheated steam flow if two boilers are connected to a common main steam line, but to separate auxiliary steam lines. Increasing the furnace heat too rapidly on increase in steam demand, or failure to secure burners quickly enough on sudden reduction of steam demand also can cause superheater tube bursts. In all cases, the overheat must be extremely sudden and severe or the alloyed superheater tubes will crack or tear before they form thin-lipped bursts.

Thick-lipped bursts resemble thin-lipped bursts except that they generally occur with much less expansion of tube. They have thick, ragged lips as though the metal tore apart before it reached the temperature necessary for smooth plastic flow. Unlike thin-lipped bursts, they can range in size from narrow tears in ¼" diameter blisters, to gaping holes several inches in length and width. Failures of this type in plain carbon steel generating tubes indicate that the

FIGURE 12.—Thick Lip Burst, 1" Generating Tube.

tube metal has been heated in the sub-critical range for a relatively long period of time. The size of the burst is a very rough indication of the temperatures attained. For example, a small tear in a tiny bulge suggests prolonged mild overheating and eventual cracking under the high circumferential stress imposed by the abnormal temperature. Large thick-lipped bursts suggests that the tube temperature was gradually elevated through the sub-critical range to a temperature which allowed plastic flow. The thick-lipped rupture in Figure 12, which resulted from low water, illustrates this type of failure. As the water level dropped, partial water starving of the generating nest caused sub-critical temperatures and thermal stress. As the water level continued to drop more serious starving caused higher temperatures, plastic flow and eventual tearing of the weakened tube along the stress lines.

Thick-lipped ruptures are more common in alloyed superheater tubes than in plain carbon steel generating tubes. Superheaters are more liable to periodic mild overheat from fluctuations in steam flow and the alloyed metal tends to tear before it reaches the temperature for smooth flow. Because of these factors, thick-lipped bursts occur over a wide range of temperatures in superheaters. Tube swelling is generally less than in the thick-lipped bursts of generating tubes.

Heat blisters are a less pronounced effect of overheat and plastic flow in the tube metal. Typical blisters are

elliptical bulges with their long axes parallel to the long axis of the tube. They range in depth and diameter from a fraction of an inch to several inches, depending on the size of the tube and the extent and duration of overheat. For a blister to form, the tube metal must be heated high enough to start plastic flow and then cooled before the blister swells enough to burst. Momentary stoppage by a foreign object can cause this effect, but water side deposits are by far the most common cause of ordinary blisters. Scale or sludge restrict heat transfer and raise the temperature of the metal to the point of plastic flow. The resulting bulge cracks the deposit from the water side and boiler water quenches the blister. Deposit may continue to accumulate over and over again in the same spot until thick-lipped bursting occurs. Figure 13 shows a typical heat blister in the water drum curve of a generating tube.

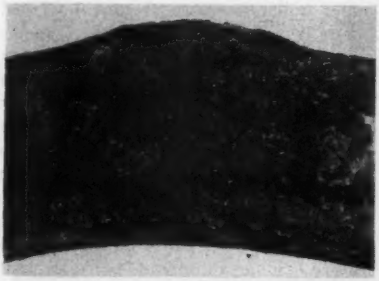

FIGURE 13.—Heat Blister, 2" Generating Tube.

Heat blisters are rare in superheater tubes because stoppage and deposits seldom occur in them. Occasionally a superheater tube will bulge by growing uniformly in diameter, so that the damaged portion is considerably larger in cross sectional area than the original tube. Very mild overheat from steam starving and gradual weakening by thermal stress promote this type of enlargement. Figure 14 shows an enlarged superheater tube.

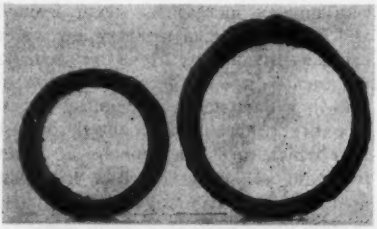

FIGURE 14.—General Swelling, 1" Superheater Tube.

Tube corrugation is a peculiar type of heat blistering which occurs when the boiler water is contaminated with oily matter. It consists of closely spaced, small diameter, exceptionally hemispherical bulges along the side of the tube facing the furnace. The pattern suggests that the tube has been heated to softening temperature and then punched from the inside with a blunt instrument. It is difficult to say why oil contamination causes this particular type of deformation. One possibility is that the oil does not form a continuous film, but separates into globules which insulate the tube metal and allow plastic flow to occur under them. Plate 15 shows a typical example of corrugation in a fire row generating tube. A surprisingly small quantity of oil on the water side will cause this effect.

Sagging is the mildest form of thermal deformation in boiler tubes. It results from heating the tube metal to a temperature where the tendency for plastic flow is just sufficient for the tube to sag under its own weight. Sagging occurs only in oblique and horizontal

FIGURE 15.—Heat Corrugation, 2" Generating Tube.

tubes. This effect, in the absence of bulging, indicates not only that the overheat was extremely mild, but that it was very uniform both longitudinally and circumferentially. Partial starving of the tube by incipient low water, or entrainment of steam in the downcomers during high power operation, can cause this effect.

Warping resembles sagging in that it consists of general distortion of the tubes. It differs from sagging in that the bending may be in any direction. Warping can result from purely mechanical stresses, such as tube explosions. However, thermal stress during rapid cooling is the most common cause of serious warpage. No overheating of the tube nest is necessary to cause this effect, and the tube metal generally will not show evidence of overheat when examined in the laboratory. This fact and the tendency of sagging always to occur in a vertical direction makes it reasonably easy to distinguish between warping and sagging.

Material deformities are relatively rare in boiler tubes. Dents and folds sometimes are formed during the shaping of curved boiler tubes. Likewise, drawn tubes are sometimes neck down so that they appear to be externally corroded on first examination. A fold can be distinguished from a heat blister by the fact that its long axis is located transversely whereas a heat blister's long axis is located longitudinally. Convex and concave folds sometimes occur simultaneously on opposite sides of the same tube as shown in Figure 16. Laboratory examination is the only means of confirming the absence of temperature effects in deformations of this type.

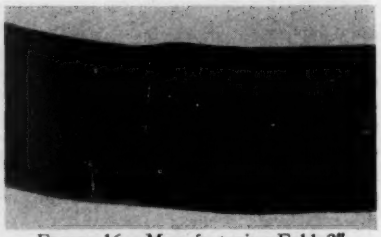

FIGURE 16.—Manufacturing Fold, 2" Generating Tube.

Tube necking can be distinguished from wasting by comparing the external and internal diameters of the tube. In mechanical necking, both diameters are distorted in the same fashion, while in corrosion necking, the exterior is reduced in diameter, while the interior remains the same. Figure 17 shows a mechanically necked superheater tube.

FIGURE 17—Mechanical Necking, 1" Superheater Tube.

CRACKS AND FISSURES

This classification includes all failures which occur as narrow slits or crevices without significant deformation of tube shape or loss of metal. It includes the many types of thermal cracks, mechanical cracks, corrosion fatigue fissures, embrittlement fissures and manufacturing imperfections, which do not readily fall into the two classifications previously described.

Creep cracks are the most common example of this type of failure in boiler tubes and superheater tubes. These generally are closely spaced, narrow longitudinal, straight-walled splits in the tube metal. They occur on either the water side or the fire side of the tube wall nearest the flame and vary in depth from superficial cracks to complete penetration of the tube wall. They result from maintaining the tube at sub-critical temperature for a relatively long period of time, so that the increased circumferential stress (hoop stress) gradually tears the metal apart. If this condition persists long enough, corrosion may set in the cracked areas and convert them to corrosion fatigue fissures. Likewise prolonged stress can develop creep cracks to the point that the weakened tube bulges and breaks in a thick-lipped burst. Higher temperatures are required to cause creep cracking in alloyed superheater tubes than in plain carbon tubes; however, the appearance is essentially the same in all cases. Figure 18 shows creep cracks in a generating tube which resulted from partial stoppage.

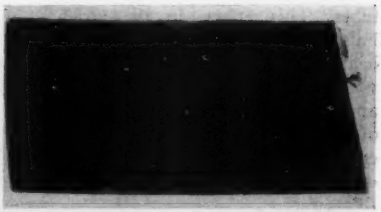

FIGURE 18.—Creep Crack, 1" Generating Tube.

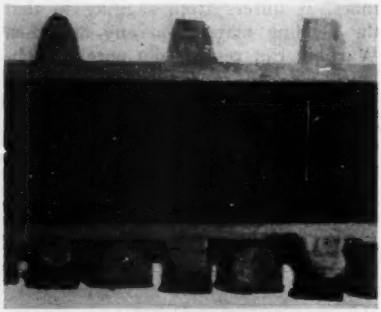

FIGURE 19.—Corrosion Fatigue Fissure, 1½" Tube.

Corrosion fatigue fissures are closely related to stress cracks in that corrosion proceeds very rapidly in stressed areas. As corrosion proceeds, stress has a more pronounced effect because of reduction in a wall thickness and a vicious cycle is established. A typical example of stress corrosion is shown in Figure 19. This is a water side crack in a wall tube directly behind a soot blower support lug. Figure 20 shows an enlarged cross section through the crack and the lug. The one sided weld caused vibration in the tube wall, which stressed it and allowed corrosion fatigue to set in. When examined microscopically, corrosion fatigue fissures differ from tube cracks in that they have conical, cavitated walls rather than angular crystalline walls. In most cases corrosion does not follow grain boundaries, but is transcrystalline. Operators and maintenance men cannot be expected to diagnose corrosion fatigue failures accurately. In most cases they should be able to distinguish them from creep cracks and mechanical cracks by looking for the cavitated appearance of the break, as contrasted with the sharp crystalline appearance of a true crack.

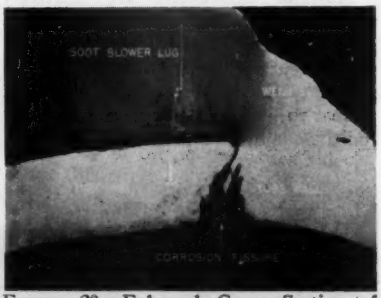

FIGURE 20.—Enlarged Cross Section of Fissure.

In high alloy tubes, like 18-8 chrome-nickel, corrosion resistant superheater

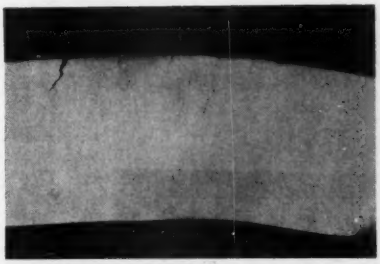

FIGURE 21.—Embrittled Steel, 5X.

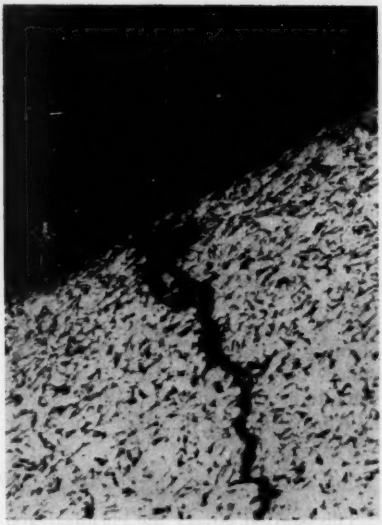

FIGURE 22.—Embrittlement Crack, 100X.

tubes, cracks sometimes occur transversely, (circumferentially) instead of longitudinally. These are brought on by very mild overheat. They result from precipitation of carbide at the grain boundaries and resulting fracture of the metal along the crystal interfaces. These cracks usually represent a lower degree of overheat than the longitudinal creep cracks.

Embrittlement cracks are a very special type of corrosion fatigue fissure. They cannot be detected by a casual examination of the metal. However, when the metal fails under pressure, or is bent or twisted, it breaks with irregular, clean, bright, crystalline edges. Embrittlement results from concentration of boiler water caustics in crevices in strained areas, such as riveted drum seams. The caustic is just corrosive enough to dissolve the grain boundaries of the strained metal, but not the grains themselves. For this reason it travels along the grain boundaries and produces extremely narrow, meandering voids which allow the metal to tear apart when it is severely stressed. Figures 21 and 22 show an embrittled sample of steel and a magnified view of one of the embrittlement cracks. This type of damage is virtually non-existent in Naval boilers because of the use of welded seams and the tendency of chloride in the sea water distillate to prevent concentration of caustic. It occurs occasionally in shore station boilers and locomotive boilers. In any case, embrittlement can be proved only by laboratory examination.

Mechanical fatigue cracks appear as clean, crystalline breaks in the metal, and in this respect look very much like creep cracks, corrosion fatigue and embrittlement on first inspection. They generally show brighter fractures than other cracks. If examined microscopically, they will not show the high temperature characteristics of creep cracks, the intergranular pattern of embrittlement cracks or the corroded edges of corrosion fatigue fissures. True mechanical fatigue cracks rarely occur in boiler tubes because they tend to promote corrosion fatigue. For example, the corrosion fatigue failure illustrated in Plate 20 would have been a mechanical fatigue failure if it had not been for the effect of the boiler water on the mechanically stressed metal. The circumferential breaking of superheat tubes near their headers probably is a combined effect of mechanical fatigue and

creep failure. Laboratory examination is required to distinguish between these classes of cracking.

Manufacturing faults account for some crevices and scores in boiler tubes. A typical example is shown in Figure 23. This score on the exterior of the tube resulted from a slag inclusion. Originally it existed as a longitudinal hairline mark along a considerable portion of the tube's length, with short, transverse hairline cracks approximately ¼" apart. When put into service the slag insulation allowed the outer shell of metal to burn away and produce the wider fissure shown in this figure. Longitudinal hairline scores or seams also are found on the insides of boiler tubes. These have a tendency to promote corrosion. Manufacturing faults generally can be identified by the high degree of uniformity that characterizes machine fabrication. They are a very remote source of boiler tube failure.

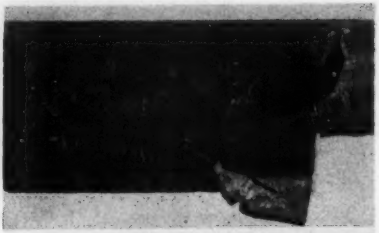

FIGURE 23.—Manufacturing Score, 1½" Tube.

BURNS

This section covers the chemical and physical effects of high temperature on the tube metal, aside from the deformation and cracking discussed earlier. Generally speaking, burned tube metal is black instead of the characteristic water side reddish-brown or fire side reddish-black. Burning can occur on either side of the tube with a variety of results.

FIGURE 24.—Fireside Burning, 1" Generating Tube.

Fire-side burning produces a brittle, scaly, black coating by reaction of oxygen with the metal to form black oxide. This reaction starts in carbon steel tubes when the temperature reaches about 1000°F. It can occur without serious burning of the water side if the water side is protected by a deposit. Bulging or bursting need not accompany fireside burning, but some sagging or warping almost always results from it. Fireside burning causes permanent stress damage in the tube metal and weakens the tube wall by converting it to brittle oxide. Figure 24 is an example of external burning. The most serious cases occur in connection with heat blisters and low water.

Water side and steam side burning start with formation of a thin film of black, shiny, magnetic iron oxide. This can form either in a clean tube or under a deposit. It results from the reaction of steam and iron at temperatures above 750°F in plain carbon steel tubes and above 1000°F in regular alloyed superheater tubes. Prolonged high temperature increases the oxide to thick flaky layers and reduces the metal thickness correspondingly. Heavy oxide films have a tendency to crack in haphazard patterns on cooling. Tube stoppage and low water are the most common causes of severe waterside burning in generating tubes. In superheater tubes the causes are, reversal of steam flow, overfiring the superheater at low loads, overfiring the saturated side of a divided fire box boiler without steam flowing in the superheater and tube

stoppage. Burning generally results in other effects, such as creep cracking, blistering or bursting. Figure 25 shows a burned waterside.

FIGURE 25.—Water Side Burning, 1" Generating Tube.

Melting is the most serious type of overheat. It occurs when there is no water or steam in the tube. The temperature increases until the metal either melts and drips away or burns completely to iron oxide. In plain carbon steel tubes, melting starts at about 2700°F. This is very close to maximum furnace temperature. Serious burning of this type can be recognized by the porous, silvery black appearance of the remaining metal and by the presence of fused globules of metal. Figure 26 shows melting of generating tubes.

FIGURE 26.—Melting, 1" Generating Tubes.

Steam tracking is a relatively uncommon type of water side burning which occurs in generating tubes. It appears as a narrow band of black oxide in an otherwise normal tube. The tube wall nearest the flame is the one affected. Steam tracking is most common in horizontal floor tubes and oblique roof tubes, but its occurrence in vertical tubes is possible. Figure 27 shows a steam track on the top wall of a floor tube of an experimental 2000 psi shipboard boiler. The narrow band of black oxide indicates that faulty circulation allowed formation of a continuous steam film along the line of contact with the flame. In this case the insulating steam film gradually burned the metal away from the inside and produced thermal stress and eventual creep cracking. Figure 28 is a cross section of the failed area. The creep crack and the loss of metal from the water side are evident. This type of failure can take place in a steam track even though the remainder of the tube is completely normal. Improvement of circulation by increasing the pumping rate in forced circulation boilers, or by changing the tube angle in natural circulation boilers are the only means of overcoming steam tracking.

FIGURE 27.—Steam Track, 1" Generating Tube.

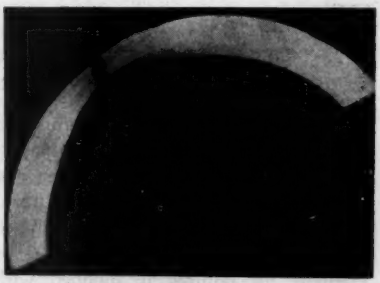

FIGURE 28.—Enlarged Cross Section of Track.

SELECTION OF TUBE SAMPLES

The information which can be obtained by laboratory examination of tube failures depends to a large extent on the care taken in removing and marking the specimens. The basic rules are: that the specimen contain a good example of the actual failure, that it be removed in such a manner that the failure will not be disfigured, and that it be marked to show its original position in the tube nest. A tube with a crack or hole near the tube sheet or header should not be cut out with a torch which will ruin the specimen. Likewise, a tube should not be sawed with oily blades which might ruin waterside deposits. It is preferable to mark each specimen by die punching to show its furnace side and its header or drum end, as well as its location in respect to adjacent sections of the same tube and other tubes in the nest. It is helpful to circle failed areas with chalk or grease pencil if they are difficult to find. In cases of tube deformation, burning and cracking, sample tubes should be taken on both sides of the failed tube so that the laboratory can determine if the effect was local or general. In no case should tubes be cleaned chemically, sandblasted or flattened before they are sent to the laboratory for examination of failed areas.

Importance of Terminology in Casualty Reports. Better understanding of boiler casualties can be promoted by eliminating ambiguous terms from casualty reports, including reports from the technical laboratories. Descriptions of failures should indicate clearly whether the damage appears to have resulted from cracking, corrosion, burning, blistering or bursting. In cases of cracking and corrosion it should express an opinion as to whether the damage started on the inside or the outside of the tube. In cases of pits and localized, shallow cracks this can be determined easily. In cases of general corrosion, internal wasting can be distinguished from external wasting by comparing inside and outside diameters with those of an unfailed area in the same tube. The use of general terms, such as perforation, for describing all types of holes in boiler tubes can be very misleading. If a thin-lipped or thick-lipped rupture is called a burst, there will be no misunderstanding. Similarly if a localized hole is called a pit, or a narrow split is called a crack, fissure or split, the reader will have a reasonably good picture of the type of failure. General loss of metal over a large area on either the fire or water side can be properly described as wasting. The terms blister and bulge are satisfactory for describing enlarged, unruptured areas.

CONCLUSIONS

By looking for the distinguishing characteristics discussed in this paper, operating engineers and repairmen can accurately diagnose a majority of naval boiler tube failures. A properly equipped laboratory can determine the cause of virtually all failures if supplied with representative tube specimens and proper background information. Care in sampling and tagging specimens and use of proper terminology in describing damage will lead to a better understanding of tube failures and to gradual elimination of their principal causes.

TWO THOUSAND YEARS OF ENGINEERING

ACKNOWLEDGMENT

This article is an abridged version of SIR CLAUDE GIBB'S presidential address to Section G (Engineering) of the British Association for the Advancement of Science, delivered in Edinburgh on August 10, 1951. The abridged version was printed in the August 17, 1951, issue of "Engineering."

Engineering is much older than any history that has come down to us, for it stretches back to the beginning of civilization. With the growth of the first civilized countries, problems arose in the cultivation of the land, and in transport. Engineers were called upon to provide irrigation schemes, reservoirs, canals, dock and harbors. The rulers of the day required them to build palaces and great monuments, examples of which are the Pyramids. Temples had to be erected. These presented great problems for the engineer. To shape the large blocks of stone used for their construction, tools had to be designed and built, and here again, the engineers displayed much skill and ingenuity.

During the Third and Second Centuries B.C., Greece furnished a number of eminent engineers, records of whose engineering feats have been passed down to us. Ctesibius Alexandrinus invented a two-cylinder force pump; the remains of one are in the British Museum. Hero of Alexandria was possibly a follower of Ctesibius, and his researches brought him into the fields of mechanics, catoptrics, pneumatics, surveying, geometry and stereometry. By the engineer he is remembered for his original idea of a reaction turbine, and also for his hydraulics. The modern surveyor owes his theodolite to Hero, who called his own instrument the dioptra. Archimedes was another Greek genius; he laid the foundation for our knowledge of specific gravity, and of the principles of leverage and the water screw. The mathematical problems which he solved are of vital importance to the engineer today.

The development of the Roman Empire set the engineers great tasks. The greatest achievement of Roman technical work was the making of roads. With the decline of the Roman Empire, little progress in engineering was made until 1629, when Giovanni Branca, of Rome, published information on a new steam engine. This was, in fact, a steam turbine which operated on the impulse principle.

About this time, coal mines in Great Britain were becoming deeper, and difficulties were being experienced in pumping out water, and raising the coal. Engineers turned their attention to the design of an engine for this purpose, and in 1663, the Marquis of Worcester described a method of raising water by the use of steam. During the next century, the reciprocating piston engine made its appearance, and the names of Papin,

Savery, Newcomen, Beighton, Smeaton and Watt will always be associated with its development. Progress in all branches of engineering, from then on, developed rapidly. Steam engines were fitted into river boats in this country, in America and in France. In 1819, the *Savannah*, a vessel of 300 tons, fitted with a single-cylinder steam engine driving paddle wheels, crossed the Atlantic from America to Europe: but some years had to pass before vessels and machinery of adequate strength and power were available to inaugurate the transatlantic service.

Henry Cort had discovered in 1783 a process for producing wrought iron by puddling, and rolling it into plates and bars, an invention which provided a new material for engineers. The history of iron ships began in 1787 with an iron canal barge. The first iron steam vessel was the *Aaron Manby*, which David Napier navigated from the Thames to the Seine in 1822. Screw propulsion was also introduced about this time. Engineers were also concentrating on improved land transportation. Trevithick, Blenkinsop, Hedley, and George Stephenson will be remembered for their work in constructing steam locomotives for coal haulage. Stephenson also constructed the locomotive which drew the first public passenger train in the world, on the Stockton and Darlington Railway, in 1825.

These developments in the steam engine had been made possible by the work of the engineers and physicists who did preliminary work on determining the laws of steam. Watt discovered that the latent heat of steam is nearly constant for any pressure within the range of steam-engine practice, and that, consequently, the greater the steam pressure, and the greater the range of expansion, the greater will be the work obtained from a given weight of steam. He also invented the steam-engine indicator. Count Rumford carried out experiments on frictional heat, and, in 1824, Sadi Carnot published his pamphlet which gave us the pressure-temperature cycle. Carnot's ideal diagrams, with isothermal and adiabatic lines, represented very closely the action of the steam in engines of that period, but he failed to recognize that heat disappeared in the process of producing power, in an amount equivalent to the external work done.

Joule, in 1849, established the indestructibility of energy, the mechanical equivalent of heat and the existence of the absolute zero of temperature. In the following year, Sir William Thomson (afterwards Lord Kelvin) deduced the absolute zero of temperature from Carnot's cycle and reached the same conclusion as Joule. Thomson, Rankine and Clausius continued the work by giving the subject mathematical treatment and it is to these physicists that we are indebted for the discovery of the laws of thermodynamics, and to Clausius for the introduction of the term "entropy."

The beginning of the Nineteenth Century was also an important period in the history of electricity. Although magnetism had been known since ancient times by the Chinese and the Greeks, little use was made of it. One of the first references to its use for practical purposes occurs in the description of Columbus's first voyage to America in 1492, when he used a compass. Little was achieved during the next 300 years until Volta, discovered, in 1799, that electricity could be obtained by chemical action, and produced the voltaic cell giving a steady low-tension current. Rapid progress in electrical research was thus made possible, resulting in discoveries by Oersted, Ampère, Faraday, Davy, Sturgeon, Ohm, Henry and finally Faraday again in 1831, when he made the greatest contribution to the development of electrical science by constructing the first magneto-electric machine. Further developments were made by other inventors, and, by 1844, one of the first effective industrial electric generators was constructed for an electro-

plating works at Birmingham. Another field of development was that of the telegraph. Attempts had been made in the Eighteenth Century to introduce the electric telegraph, but it was not until 1837 that success was achieved independently in the United States by Samuel Morse, and in England by Wheatstone and Cooke, who installed, in 1843, the first public electric telegraph line, between Paddington and Slough.

By 1850, reciprocating steam engines were in general use both on land and sea; the preliminary laws of thermodynamics had been established; the steam locomotive was in use and passenger train services were being inaugurated in different parts of the country; important developments had been made in electricity; electric generators had been constructed and the electric telegraph services were in use; wrought iron had been discovered and was being used for the construction of boilers, steam engines, ships and bridges.

With the success of the electric telegraph on land, the first submarine cable was laid in 1850 between Dover and Calais; and, after many attempts, a transatlantic telegraph service was established in 1866 between Ireland and Newfoundland. It was operated by the Atlantic Telegraph Company. In the early days, the transmitting speed was 15 letters per minute, since increased to over 2500 letters per minute. The transmission of speech was next attempted, though it was not till 1876 that a practical telephone was invented by Alexander Graham Bell. At the British Association meeting held at Glasgow, in that year, Graham Bell, in conjunction with Lord Kelvin, brought it to the Association's notice. Within a few years, telephones were in use in all the advanced countries in the world.

Even more revolutionary was the transmission of power by electricity. The first proposals for electrical transmission concerned electric lighting, and it was this development that led the way to the transmission of power. The electric arc-light, first demonstrated by Humphry Davy in 1809, had been turned into a practical commercial possibility by Foucault in 1844, but, while it was used for lighting railway stations, docks, etc., it was not ideal for indoor lighting.

Many inventors attacked the problem of making an incandescent electric lamp, but without success, and it was not till 1878 that Sir Joseph Wilson Swan made such a lamp and exhibited it at Newcastle. T. A. Edison was also working on the problem and he patented his process in 1879. Companies formed by the two inventors were amalgamated in 1885 to form the Edison and Swan United Electric Light Company. The first opportunity given to the public of seeing electric incandescent lighting on a large scale was at the Crystal Palace Electrical Exhibition, 1885.

Edison saw that the possibilities of electric lighting were dependent upon the efficient supply of current from a generating station, and set himself the task of designing a reliable dynamo. In 1879, he produced a dynamo to give electric current at a constant pressure of 110 volts. These dynamos were designed to be driven either by a pulley and belt, by a steam engine, or by coupling the armature shaft directly to the crankshaft of a steam engine. Edison also worked out the (by no means obvious) system of distribution by cable and wiring circuits.

The first public lighting supply was opened at Appleton, Wisconsin, in 1881, and in the same year the first station in England was opened at Godalming, in Surrey. In the following year, the much larger Pearl Street station, New York, was opened with six generators, with an eventual lamp connection of 6000 to 7000 lamps. By the end of the year, nearly 80 electric companies were

founded in England, though few had actually gone into operation. In 1883, the first public electric railway was opened, between Portrush and Giant's Causeway in Ireland.

When central-station electric lighting by direct current was first established, it was found that there were many towns and places where the demand was so scattered that it did not pay to put down the heavy feeders and distribution mains required on the direct-current system. Attention was therefore directed to the possibilities of alternating-current supply. Many different types of alternators were designed by Gramme, Westinghouse, Ferranti, Kapp, Parker, Mordey, and Messrs. Ganz of Budapest. Ferranti especially devoted his attention to the design and construction of large alternators for public electric supply, and installed a number of them in power stations in London. The Deptford station will always be associated with the name of Ferranti, for it was there that he gave a lead to the world by installing single units of 10,000 h.p., generating at the unheard-of pressure of 10,000 volts.

In the early days of electric lighting, it was usual for the generating unit to be driven by a belt from some kind of steam engine. Semi-portable engines of the agricultural type were generally employed, and there was a serious need for some form of high-speed engine for direct drive. One of the earliest high-speed direct-coupled engines was made by Peter Brotherhood, who arranged his three-cylinder engine to drive a dynamo directly from either end of the crankshaft. This was soon superseded by the Parsons epicycloidal steam engine of 1877, which, in turn, was outclassed by the Willans central-valve engine. The latter was displaced by the Bellis double-acting high-speed compound engine.

While this evolution of the reciprocating steam engine was taking place, the final solution, namely, the steam turbine, was being brought to practical success by Sir Charles Parsons. He patented his first turbo-dynamo in 1884, and it was this little machine, of only 7¾ kw, which led the way to the large central-station turbo-alternators built today. Parsons developed the turbine chiefly in relation to electricity generation, and many small turbo-generators were built for supplying current for lighting purposes in ships, factories, hotels, etc. After the incorporation of condensers in 1891, the turbine rivalled the reciprocating engine in efficiency and was rapidly adopted for power stations. By 1912, turbo-alternators for an output of 25,000 kw had been constructed; today single units of over 200,000 kw are in operation.

The commanding position of the steam turbine is due mainly to its high thermodynamic efficiency; in large sizes, it gets more work out of coal than can be got by any other prime mover, due to its ability to take full advantage of the high pressure and high temperature steam conditions now in use, and the adoption of regenerative feed heating, in which steam is tapped from the turbine at successive stages of the expansion to heat the feed water on its way back to the boiler.

Parsons also attacked the problem of adapting the turbine to marine propulsion, and, after many disappointments, was equally successful. He constructed a small launch, the *Turbinia*, in 1894, which was fitted with a single turbine driving one propeller. On trials the speed attained did not give the expected result due to propeller cavitation. New turbines and new propellers were fitted which ultimately enabled a speed of 34 knots to be obtained. During the naval review at Spithead in 1897, the *Turbinia* was demonstrated before a fleet representing not only the British Navy but the seapower of other leading nations as well. Her performance focused the attention of the Admiralty on the possibilities of turbine propulsion, and they

entrusted Parsons with the construction of a 30-knot turbine destroyer, H.M.S. *Viper*. In the following years, the turbine was tried in various vessels of the Navy with such success that, in 1905, a Committee on Naval Design advised that turbines should be used exclusively in all classes of warships.

Meanwhile, the first turbine-driven passenger vessel, the *King Edward*, was built in 1901 for service on the Clyde; she was followed by cross-channel boats, and later by vessels for the Liverpool-Canada passenger service. In 1906, the *Lusitania* and *Mauretania* were each fitted with four turbines totalling 70,000 h.p. Today, the turbine is the recognized prime mover for all the navies of the world, as well as for all the fastest ocean liners. It made possible the construction of the *Queen Mary*, *Queen Elizabeth*, and many others. The introduction by Parsons of gearing between the turbines and the propellers enabled the size of machinery to be reduced, and increased its efficiency.

To use fuel directly in a cylinder, instead of the more usual cycle of fire, boiler and cylinder, attracted the attention of inventors from the earliest days, but the arrival of the practical steam engine put an end for a time to work in this direction. It was not till the Nineteenth Century, when coal gas became available, that inventors once again took up the subject. Coal gas provided a suitable and readily available fuel, and in 1823 Samuel Brown constructed the first commercial gas engines. Progress after this became rapid, due to the attention of many more inventors, till in 1876 Otto produced the famous "Otto silent gas engine," completing its cycle in four strokes. Two years later, Dugald Clerk produced his two-stroke engine which, for large powers, tended to be the more favored.

Although large units using blast-furnace gas were developed, with efficiencies greater than steam in suitable circumstances, today it is regarded merely as a step in the evolution of the petrol and oil engines. Petroleum in large quantities had been discovered in 1858 in the United States, and the increasingly wide distribution of petroleum oils, particularly the paraffin oils and lamp oils, focused attention on the possibility of using oils as fuel instead of gas. The earliest oil engine to achieve success was that of Priestman, introduced in 1885; followed the next year by Daimler's engine, which used an oil so volatile that a carburetor would serve to charge the incoming air with combustible vapor, and which ran at high speed, enabling bulk and weight to be lessened and power increased. The Daimler engine marks the beginning of the modern gasoline engine. Many inventors contributed to the development of the oil and petrol engine, one of the most notable being Rudolf Diesel, who, in 1895, produced an engine which, while able to use crude oil, gave a remarkably low fuel consumption and high efficiency. The Diesel engine was developed for use on heavy road-transport vehicles and, in the larger sizes, as a rival to the steam turbine for marine propulsion, where it found many advocates. Much attention has also been paid to its use on railway traction, and the number of Diesel main-line locomotives is growing rapidly.

The effects of the petrol engine were most revolutionary in the field of transport, first on the roads and then in the air; its lightness in relation to power made it pre-eminently suitable for this purpose. Daimler at first intended his engine to be used for stationary work and for the propulsion of boats, but in 1886 he fitted it to a bicycle. The results were so successful that, three years later, he built another engine for a road vehicle. This was the beginning of the great motor industry. The year 1896 saw the founding of the English Daimler Motor Company, and in the same year the Ford Motor Exhibition was held in the Crystal Palace, London.

These early cars were frequently unreliable, but the mechanical design was quickly improved as a result of practical experience on the road. Other improvements, such as better systems of ignition and cylinder cooling, pneumatic tires, and streamline body construction have done much to popularize the motor car. By 1928, the output of high-speed internal-combustion engines exceeded by more than ten times the total horsepower of all power stations, ships and railways. Again, at the World Power Conference in Berlin in 1930, it was stated that the number of motor cars on the world's roads was some 30 millions, with an output of at least 600 million h.p. Today, self-propelled vehicles are used for practically all forms of transport.

One of the most notable applications of the petrol engine is in aviation. It was not till the Eighteenth Century that flight was achieved by the use of hot-air balloons. Thereafter considerable experimental work was undertaken with power-driven model airplanes, the first model to rise under its own power and land safely being constructed in 1857. With the rise of the motor-car industry and the availability of the light and reliable prime mover, attention was directed to the possibility of controlled power flight.

The Wright brothers, who had been carrying out extensive gliding experiments, designed a motor and propeller to suit their machine, and in December, 1903, were able to rise in the air and fly for a distance of 25 yards. This was the first flight of a machine fully controlled. Progress was rapid, and successful machines were constructed in 1909 in England by Cody and by Roe. In the same year, the world was startled by the first successful flight across the Channel by a Frenchman, Louis Bleriot. With the advent of World War I, technicians, factories and Government funds were made available for urgent development work and a huge British aircraft industry was built up in an incredibly short time. Between 1914 and 1918, Great Britain manufactured over 50,000 airplanes, most of the engines for which were constructed in this country. The reliability of the airplane was demonstrated by the flight across the Atlantic, in 1919, of Sir Arthur Brown and Sir John Alcock. Soon afterwards, regular air services were introduced between London and Paris. Between the two world wars, much research and experimental work was undertaken, particularly by Air Commodore Sir Frank Whittle, on jet-propulsion gas turbines for aircraft, and in May, 1941, the first airplane fitted with such an engine made its first flight.

Probably the first working gas turbine was constructed by Armengaud and Lemale in Paris in 1904, and a second machine was constructed by Brown Boveri and Company in 1906. Thereafter, gas turbines were used merely as auxiliaries or as a convenient source of obtaining power from otherwise waste energy, and not as serious competitors for power generation. Serious consideration was again given to the subject about 1935, when many steam-turbine manufacturers began investigations. The war slowed up these developments, but, as a result of the progress made in the use of gas turbines for aircraft, manufacturers in this country and abroad turned their attention again to the possibility of building successful industrial gas turbines. A number of firms have built experimental gas-turbine units from which important data have been obtained. On the Continent, gas turbines for outputs up to 27,000 kw are in operation, and in this country machines for outputs up to 15,000 kw are nearing completion. During 1950, gas turbines were used as prime movers to drive a motor car, a motor launch and locomotives.

The first idea of wireless communication was suggested by James Bowman Lindsay, of Dundee, in 1845, who pro-

posed to use the earth or sea as a conductor, but on account of the very weak electric currents involved and the lack of sufficiently sensitive apparatus to detect them, this means of communication made little progress. During the latter part of the Nineteenth Century, several British scientists conducted experimental research relating to radio-telegraphy, notably Clerk Maxwell, Sir Oliver Lodge, Professor D. E. Hughes and Admiral Sir Henry Jackson. It is, however, to Senator Marconi that we are indebted for his inventions in 1895, which made long-distance wireless telegraphy possible. He conducted his original experiments in Italy, and in 1896 came to England. By 1899 he had established wireless communication between England and France, and two years later had bridged the Atlantic. A regular transatlantic telegraph service was established during the next seven years.

Following on the success of the wireless telegraph, experiments were commenced almost immediately on the possibility of radio telephony. The most important inventions relating to this subject were, perhaps, the thermionic valve invented by Dr. J. A. Fleming in 1904, and the three-electrode tube invented by Dr. Lee de Forest in 1906. During 1914-1918, radio-telephony was used widely by the fighting forces, and valve techniques were developed rapidly to enable communication to be made with aircraft. By 1919, important experiments had been conducted by the Marconi Company and telephonic speech was established between the United States and Paris, and between Ireland and Canada. In the same year, experimental broadcasting transmissions were commenced in England, which culminated in the forming of the British Broadcasting Company in October, 1922, and the setting up of regional stations.

The first details of a method to transmit actual images was disclosed in 1880 by Carey, an American, but it was not until 1926 that the most successful proposal was described by John L. Baird. Baird used a scanning disc which carried two spirals of lenses, and the reflected light from the illuminated object was caused to affect a photo-electric cell. In 1929, the B.B.C. and the Baird Television Company commenced a public television broadcasting service from Alexandra Palace, London. It is expected that television broadcasting programs will be available throughout the country by 1952.

During recent years, considerable attention has been directed to the work of the scientists investigating nuclear energy, and to the possibilities of adapting this new source of energy to commercial work. The beginning of this era goes back to 1789, when a German scientist, Martin Heinrich Klaproth, discovered a substance which he named uranium. Over 100 years elapsed before its peculiar properties were observed by Wilhelm Röntgen, who discovered X-rays, but it was not till 1896 that Becquerel identified the phenomena as radio-activity. The release of this information aroused interest in other scientists, notably Professor Curie and his wife Marie Curie, and G. C. Schmidt. These scientists, working independently, discovered radioactive properties in thorium, polonium and radium, and Debierne discovered, in 1899, actinium. Chiefly because of the work done by Lord Rutherford, we know that these substances are spontaneously disintegrating into a series of radioactive substances, each of which is of lower atomic weight than its parent element. Many other scientists contributed to our knowledge of radio-activity, but it is to Lord Rutherford that we are indebted for much of the pioneer work leading to the release of nuclear energy.

In 1942, the first atomic pile or nuclear reactor was put into operation in America and the first man-controlled

release of atomic energy was achieved. During the war, considerable progress was made due to the concentration of scientific personnel, and the spending of sums of money which, in peace time, would have seemed enormous. Since 1945, atomic energy research establishments have been put into operation in this country; at Harwell, under the direction of Sir John Cockcroft, and at Sellafield under Sir Christopher Hinton.

And now 1951. We have seen the evolution of the steam turbine and its use on land and sea; the turbo-alternator for the generation of electricity; the distribution of electricity to factories and homes, and the manufacture of the thousands of miscellaneous electrical appliances; the internal-combustion engine and its use on land, sea, and in the air; the telegraph service; the telephone; wireless broadcasting; television; the gas turbine; the motor car; the airplane and finally nuclear energy. But in addition to these outstanding developments there have been many thousands of inventions relating to almost every branch of industry, such as machine tools, the typewriter, calculating machines, the gramophone, the bicycle, and the mechanization of coal mining and agriculture.

There seems little doubt that a tendency in the future will be towards large-scale railway electrification where it can be carried out economically. This tendency may well be hastened by the application of nuclear energy, on a large scale to the production of electrical energy. The production of electrical energy from fissile material is, even in the light of our limited experience today, a practical possibility. With the prevailing costs of coal and petroleum, and of uranium or other fissile materials, it is doubtful whether electricity could be produced, at the moment more economically than by well-established methods; yet it requires but small improvement in the method of using the heat generated by fission to enable atomic energy to compete with present-day practice. To justify the high initial cost of an atomic energy plant the production of electrical energy by fission must be carried out on a large scale. Electrification of railways and other large-scale usage of electricity will automatically result.

For a long time now, I have advocated the rational usage of our greatest national asset—coal. It is unlikely that, in the future, our available supplies of coal will increase; on the contrary, they are likely to become less year by year. Economics will therefore force us to use coal more economically and I foresee high-pressure gas mains supplying heat to home or industry, the complete abolition of the open coal fire, the increased use of slow-combustion heating in homes, using coke or other residual fuels, gas turbines living up to their names and using gas, and the use of coke, gas or tar, and of course, high-ash coal residues for generating electricity in central power stations.

Work is now being undertaken in this country in burning methane, which occurs in small proportions in the large volumes of air used in mine ventilation and it may be that, in the future, by using a high degree of pre-heat prior to combustion, the miner's greatest enemy, firedamp, will be used for the benefit of man.

Without any doubt, in my opinion, the jet or gas turbine-cum-jet will be used to the exclusion of all other means of propulsion for all types of aircraft. This type of prime mover meets the requirements so admirably that it is difficult to see how any other form of power unit can compete. On land, the gas turbine is more sorely pressed by its competitors and I doubt if any prime mover other than the steam turbine will be used for very large powers. I do think, however, that the gas turbine will, to a great extent, supplant the Diesel engine. There

is no difficulty in making gas turbine units of 100 h.p. which could easily be fitted to motor cars within the space now occupied by petrol or Diesel engines. The major remaining problem requiring solution to enable the gas turbine to take its place in our everyday life is that of a compact, low-cost and highly efficient regenerative heat exchanger. The consideration given to this problem during our present meetings is an indication of its importance and likely solution. Once a purely rotating prime mover has been found suitable for a given application, it always becomes possible to supplant an alternative reciprocating type, and there is little doubt that, within the next 50 years, the gas-turbine motor car will be used to the exclusion of all others.

Man's standards of living and comfort have increased only as he has made mechanical power his servant in increasing degree. Electricity provides the most convenient way of supplying that mechanical power, and, if the nations of the world are listed in the order of electricity consumption per head of population, the list will be found to coincide with the nations listed in the order of living standards. Incidentally, nations listed in the order of protein percentage in their daily diet gives again the same sequence. Inevitably, as nation after nation strives for improved standards of living and comfort, there must be a tremendous expansion in the use of electricity. A consumption per head in Great Britain of five times our present usage still would not produce demand saturation.

Large-scale use of electricity for pump-produced rain and for soil heating, as a means of combating in some degree the vagaries of weather, thus helping to solve our ever-growing food shortages, is certain to come. Supersonic-frequency electronically-produced vibrations will become an everyday thing in our industrial, domestic and medical life. The thermionic valve will become an increasingly integral part of our life between 1951 and 2051.

CALCULATIONS AND DESIGN OF SLEEVE BEARINGS

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The original Figures were not reproducible. Figures 1 to 12 and 25 to 35 are not reproduced herein. Figures 13 to 24 were selected by the Department of Diesel and Internal Combustion Engines, North Carolina State College, as essential to presentation of the paper and that department kindly redrew these Figures and made them available to the Society.

Today we still find bearings which, in regard to both capacity and their material bulk, by no means satisfy the present requirements. In the following it is shown how through a logical method of analysis, and without the necessity for materials of higher quality, the load capacity can be raised and, through reduction in dimensions, material can be saved. Data on the various characteristics are given for the bearing designer (1).

THE NECESSITY FOR AND THE PROBLEM OF PROGRESSIVE BEARING DESIGN

When we check the present usual designs of bearings, it is evident that further development has been done only in a very few fields of application. It appears that a further development should be faced as generally necessary, if only in certain fields—certainly most of all in engine design, where the bearing pressures are particularly high and are becoming still higher. On the other hand, it must be emphasized that methods for increasing the load carrying capacity should not be confined to those bearings in which the presently used materials no longer suffice for higher loads. Further, we can speak of improving the capacity of a bearing if the bearing is redesigned on the basis of a correct theory without changing the presently used materials—of course only insofar as they are not priority metals—thereby changing the essential dimensions and saving materials. That such bearings, designed properly with respect to lubrication theory, are characterized by greater reliability and longer life is in the end also advantageous.

If we reflect on these things we will not shut ourselves off from the belief that a redesign, according to modern points of view, is necessary not only for highest capacity bearings but for all bearings, in regard to the very important feature of saving of material.

The comparison of a few bearing shells in Fig. 1 illustrates clearly the difference between a modern efficient bearing and the older, usual type. On the left there are bearings pictured which, unfortunately, are still met in machine design, with oil grooves in the load carrying surface. On the right are bearing shells without grooves (2) which are characteristic for their load carrying capacity. Most striking is the difference in the bearing length-diameter ratio (3). The bearing on the right is not built with a reduced seating area for the reason of scantiness but because the load carrying capacity is greater with the low L/D ratio than in the case of the bearing shown on the left. We see here a characteristic of the L/D ratio, which exerts a very material influence on the capacity and on the heating of the bearing. This is almost entirely neglected in the commonly used mode of calculations, and even in the technical literature has not always found its correct appreciation (4). The bearing length-diameter relation is but one of the many data factors on which the performance of a bearing depends. The objective in the following paper will be to show the correlation of the essential factors and to develop the criteria for the calculation and design of bearings in which the latest knowledge in the field of bearing research is applied.

The task of the bearing designer is that the bearing shall be so calculated and so designed that it can run without attention with fluid friction. Shear friction of a fluid is characterized by a state where both surfaces are separated completely, one from the other, by a film of moving fluid; thus one surface floats on the other. The whole resistance to motion results from the internal friction in the fluid and is very small in comparison with body friction. The load is transmitted from one sliding surface to the other solely by the lubricating substance through the fluid pressure phenomenon. The fluid friction is due to the little opposition to movement and the consequent little heat generation, as well as a lack of wear, the ideal to be striven for. In contrast to other ideal cases, fluid friction is practicable and attainable and its realization is the task of the designer.

THE LUBRICATION PROCESS WITH FLOATING FRICTION

That in the lubricating film a pressure can arise which is capable of completely separating the sliding surfaces with pressure loading of even more than 100 kg/cm² is brought about by two properties of the lubricant: viscosity and adhesion to the sliding surfaces. These two qualities are called, appropriately enough, the "load carrying factors" (5) of a lubricating oil as distinguished from the "life factors." Among these latter qualities are listed the resistance to deterioration due to age, the resistance to oxidation, decomposition, gumming, etc. In the case of bearing design, only the load carrying factors play a role—in particular, viscosity.

The lubricating phenomenon, in the case of floating friction, will be shown shortly in the simplest case—a pair of sliding surfaces which are plane and smooth (Figs. 2-4). In the first case (Fig. 2), the surface and the plate are parallel and the plate is moved with the velocity U. The lubricant passes without pressure through the gap under the plate. No pressure is exerted on the plate, which therefore cannot be loaded. In this case, the hydrodynamic load carrying capacity of the lubricating film is zero.

In the second case (Fig. 3), the parallel plate approaches the plane with the velocity U. Now the lubricant must be squeezed out from the gap. The more viscous the lubricating oil is, the greater is the resistance to its displacement. This displacement resistance

equals the load carrying capacity of the lubricating film. This film in this case can transmit a loading from the plate to the plane without directly touching it. Similar conditions exist in the lubricating oil film of the piston pin and the bushing of a cross head during the change of pressure direction.

In the third and most important case (Fig. 4), the plate is inclined toward the plane at a small angle so that the wedge of lubrication is constricted in the direction of motion. The area of the entering cross section is greater than the leaving cross section, whereby a damming action of the fluid occurs and a pressure builds up on the plate. The clearance adjusts itself so that the loading on the plate and the pressure of the oil film balance each other. The loaded plate thus floats on the film of lubricant.

The development of an oil pressure in the wedge of the lubricant is based on three conditions:

1. The wedge must be completely filled with a viscous fluid.
2. The wedge must narrow in the direction of motion.
3. There must be sliding motion.

These requirements must also be fulfilled in a bearing. In a sleeve bearing (Figs. 5 & 6), the wedge of lubricant is formed by the eccentric position of the journal, inasmuch as it is smaller than the bore by the bearing clearance. Through rotation the oil is drawn into the narrowing gap, which circumstance causes a pressure to develop in the film of the lubricant. The pressure increase commences at the cross section where the oil enters. After reaching a maximum value it decreases to zero at the narrowest cross section (h_o) and is approximately symmetrical about the pressure maximum (Fig. 5). From this point the pressure reduces to a slight negative pressure because of the opening wedge. If the quantity of oil present is not sufficient to completely fill the gap, a hollow space forms and dissolved gases may be released. These show their presence in foaming of the oil, which is naturally unacceptable. Foaming, therefore, always indicate that places with negative pressure exist.

In Fig. 6 the influence of bearing width on pressure development is shown. The lowest curve is applicable to the above mentioned hypothetical case for a bearing of infinite width in which oil cannot flow in an axial direction. For bearings of finite widths, the pressure varies along the axis approximately as a parabola.

Thus, in Fig. 6 curves are shown for the different bearing ratios, with the assumption that the oil viscosity is constant. Through this presentation it appears that the smaller the width, the smaller is the bearing load capacity. This impression is developed frequently in literature when actually the converse is true. We should not compare on the basis of equal viscosity, but to obtain logical results we must assume that only one oil is used. It follows that in a bearing of finite width a considerable portion of the heat generated in the oil film, by internal friction, is dissipated through side leakage. In a bearing of infinite width this feature of side leakage is absent. Therefore, it follows that in the bearing of infinite width a higher film temperature must be reached than is the case with those of finite width. In a film of higher temperature the viscosity is less and the capacity of the film to carry the load is consequently less.

It is also understood that decreasing the capacity by reason of this finite width is counteracted by the increasing capacity incident to the lower temperature. Now, if we were to narrow a bearing of excessive width, as discussed above, the increase in capacity as a result of decreasing the oil temperature will be greater than the decrease in capacity due to the side leakage of the oil. Thus, the narrow bearing shows

more load carrying capacity than does the wider one. Only with relatively very narrow bearings does the decrease in load carrying capacity prevail. There is obviously a bearing ratio which is characterized by a maximum load carrying ability.

We have yet to note that the pressure distribution is as shown in Fig. 6, when the geometric conformation of the journal and the bearing does not become distorted through deformation. Such deformation can have its origin in heavy load, sometimes through expansion due to heating in an unfavorable design (6). How, in such a case, the pressure distribution can be changed is shown in Fig. 7, which is based on the pressure measurements by Buske (7). If the bearing is stiff and relatively wide, the elastic deformation of the axis produces heavy pressure concentrations at the bearing ends with an awkward distortion of the curve (curve a). But, on the other hand, should the ends of the bearing be flexible, then the distortion of the bearing clearance, and that of the lubricating film, produces heavy bearing pressures in the middle of the curve (curve b). These phenomena become more prominent the wider the bearing. So, on this basis, very wide bearings show load capacities relatively constant.

In particular cases of high load capacities it has proven advantageous if the bearing axis is made not exactly cylindrical, but to conform to the curvature of the bearing it is lightly arched so that bearing running surfaces mate with the curved journal.

CONCLUSIONS FOR THE DESIGN

Location of lubricating oil duct—Since lubricating oil is conveyed to the area to be lubricated with almost no pressure or with only the smallest of pressure above atmosphere, it should be led systematically to clearance space elsewhere than in a pressure zone (8). However, unfortunately we frequently meet with the faulty opinion that the oil be led to the region of highest pressure. When this happens, the pressure in the film in this region is unavoidably reduced to the prevailing small pressure, and the film of lubricant—very essential in the transmission of pressures—disappears. Thus, the oil hole through the region of high pressure accomplishes exactly the opposite from that which lubrication, in every case, should aim: namely, the generation of a most practicable load carrying film of lubricant (9).

If the direction of the bearing load is constant, the oil channel can discharge anywhere into a region of low pressure. In divided bearings, the oil is frequently led in at the split. The edge is relieved in a suitable manner to provide an oil pocket of ample dimensions.

It is very convenient if the drilled passage of the oil lead discharges into the low pressure area of enlarging clearance. This suction effect is so strong that the oil can be sucked from the deeper and extended oil channels. From the research of Buske, this negative pressure measures up to .3 atm. This agrees with the result of an investigation, by the author, of bearings where pressures were measured of about 200 mm. Hg, .26 atms., which corresponds to an oil column of approximately 3 m. Thus, the bearing is capable of feeding itself and, to be sure, in quantity to spare, as an American investigation has shown (10). The possibilities of so simple an oil supply, it is true, are little known.

If the loading is from different directions and if the upper and lower shells are alternately loaded, then the oil must of course be led in only perpendicularly to the direction of loading. It is important to determine the direction of

loading before the location of the channel is fixed.

If the loading rotates with the shaft, then the pressure distribution in the oil film rotates also. Then there is no permanent point of no load on the bearing shell. The running surface of the bearing should therefore become interrupted neither by an oil hole nor by a distributing groove. As an example of this case, a correct method of oil lead-in is shown in Figs. 8 & 9. Here, the oil is led in to the region of low pressure, which moves with the running journal, from a long groove machined into the journal, which in turn receives the oil from a circumferential oil groove cut into the bearing shell and located outside the supporting bearing area. This example of loading is conveniently presented in the crank pin mechanism of an internal combustion engine, particularly of the radial type.

OIL GROOVES

As already stated, oil grooves and every kind of oil distribution channeling in the pressure zone are wholly detrimental. An oil groove there acts as a load-relieving device and reduces the pressure in the lubricating film to practically zero. In the place of one great pressure peak occurring with the absence of grooves, there appear two small areas of pressure (Fig. 10). The load carrying capacity of the lubricating film is considerably reduced and frequently no longer suffices to withstand the total bearing load. If an oil groove connects regions of different pressures, as in the cross grooving of Fig. 1, then a pressure equalization effect always takes place in the sense of a reduction to the lower pressure: under some conditions even the resultant pressure may be zero. Such grooving in every case disturbs the pressure distribution sought, and by all means should be avoided.

At times we find the belief advanced, as if it were correct, that lubricating oil grooves in the pressure areas are always detrimental, but only when pure floating friction prevails, which rarely is the case. If, on the contrary, mixed friction exists wherein the sliding surfaces come in contact here and there, then the oil should be brought between the surfaces by means of lubricating grooves. These observations are, in fact, false. Mentioning, without further ado, that pure floating friction can be expected in correct designs at low pressure speeds, the oil film transmits, in the case of mixed friction, the greater part of the load through hydrodynamic pressure and relieves the planes of contact from load quite substantially. However, through the oil grooves the generation of the hydrodynamic pressures is sensibly disturbed or wholly prevented. As for the rest, the oil can feed itself into the narrowest of crevices, so that surely such oil grooves are not required. For these reasons, the belief is strong that even in the case of mixed friction no oil grooves in the loaded surfaces should be applied.

If we hear the objection that bearing shells with cross grooves in the loaded surfaces are operating in great number and running satisfactorily, too, we only need to reply that the circumstance that such bearing shells are in operation does not establish the validity of the oil groove arrangement. On the contrary, it establishes in this particular case that the admittedly smaller pressure in the oil film is sufficient to absorb the bearing load and, further, that the bearing of a more correct design in this sense could have been built inherently smaller and with less expenditure of material.

Where oscillation of the journal exists cross grooves corresponding to the angular amplitude of the journal (Figs. 11 and 12) are frequently recommended,

whereby on the oscillatory motion all parts of the journal surface are wetted. Nevertheless, it should be noted that such practice in oil grooving, under certain conditions, may cause more damage than gain. Then, since in oscillatory motion a translational velocity exists which, to be sure, varies periodically, an hydrostatic pressure builds up as a result, but which in this case hardly suffices to produce floating of the journal. The lubricating effect of the oil is now dependent on its ability to lubricate under boundary friction conditions. Simultaneously, the hydrodynamic pressure of the oil diminishes the contact (metal to metal) unit loading and lubrication is accomplished thereby. Consequently, one should assure himself that nothing exists to hinder the development of the greatest possible pressure in the film of lubricating oil. This requires above all that we eliminate all oil grooves in the pressure zone, and as a practice locate the drilled oil passages to lead the oil into a zone of "no" pressure. Here the oil is accelerated into the loaded zone of the lubricating gap through the oscillatory motion with the assistance of the capillary forces.

OIL QUANTITY—THE LUBRICATING PROCESS—OIL PRESSURE

As to the amount of lubricating oil which should be supplied a bearing, each bearing should receive just as much oil as the lubricating gap can digest. The bearing needs this quantity of oil, which, by the way, is not consumed as it flows to the sides from the oil gap but, on the contrary, is again made available when needed after cooling and filtering. It follows, therefore, that the most suitable and most economical system is the circulating lubricating system.

In wick feed oiling, drop oiling, and the like, the lubricating oil can be passed only once through the lubricating gap and then flows out laterally and is not used again; indeed, generally it goes to waste. The present day practice is to confine its use to cases where other methods cannot be applied by any means. Inasmuch as for journal bearings, which undergo not just occasional rotary motion, the oil is usually supplied to the gap in insufficient quantity to maintain the gap sufficiently full, we should avoid, in new construction at least, lubricating oil wells for bearings with rotating journals, and instead specify a circulating system.

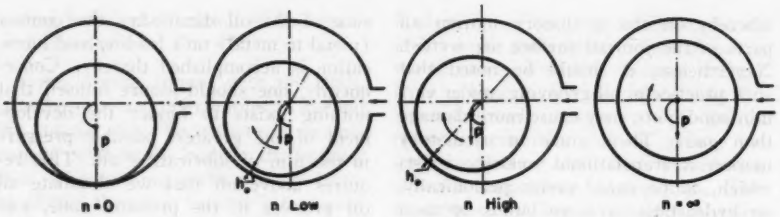
With reference to the lubricating oil pressure, the following is to be briefly noted: inasmuch as the formation of high pressures in the oil film is self-induced in the lubricating gap it is sufficient to deliver the oil without excess pressure. If to lift the oil from the sump a pump is needed, its pressure must overcome only the resistance of the pipe leads and supply channels to the flow and, in a given case, the inertia forces of oil columns in the moving parts. According to experience, 1.5 up to 3 atms will suffice amply.

The quantity of oil flowing through the lubricating gap is determined fundamentally by the rpm, bearing clearance, bearing width relations and unit surface loading. The flow of oil actually passing through the lubricating gap cannot be enlarged materially through increased pump pressure.

BEARING CALCULATION

Within the scope of this paper, which shall only give a brief summary, the mathematical derivations and details of calculation cannot be offered. We shall confine ourselves to a discussion of the basis of calculations and method of computation.

The objective of the design study is that the bearing shall be so dimensioned that it may run safely with pure floating friction,



FIGS. 13-16 POSITION OF JOURNAL IN THE BEARING AT DIFFERENT SPEEDS

h_o = Minimum Thickness of Oil Film.

The latest mode of calculation is supported by the so-called law of similitude of hydrodynamic lubrication which can be explained in a simple manner. The journal takes position within the bearing clearance always according to the operating conditions in the bearings (Figs. 13 to 16), wherein the pressure in the lubricating oil film and the unit bearing loading are held continuously in equilibrium. In a state of rest (Fig. 13) the journal occupies the lowest position. When the journal turns and the oil is drawn into the lubricating gap, a pressure results which lifts the journal and causes it to float. The higher the speed, the more oil is drawn into the gap, the higher the journal is raised and the greater is the minimum clearance. With increasing speed the journal position (its center) moves itself along a curve which, proceeding from the lowest point, approximates a semi-circle. In cases of infinitely high speeds or at zero loading, the journal center would coincide with the center of the bearing.

These nearly semi-circular curves are more precisely presented in Fig. 17. It is easily seen that the curves are not identical for all bearing width ratios. They are dependent otherwise, to a small degree, on the position of the point of oil entry and of pressure inception. These and the following presentations are valid for bearings with the oil entry

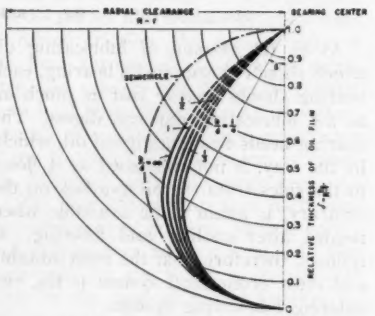


FIG. 17 PATH OF JOURNAL CENTER UNDER CHANGES OF OPERATING CONDITIONS FOR DIFFERENT WIDTH RATIOS

normal to the direction of loading, which by far is most frequently the case.

It is very advantageous to deal with dimensionless quantities into which proportional factors are introduced as relative magnitudes. The bearing clearance —namely, the difference in diameters of the journal and the bearing (D-d)—is expressed in proportion to the journal diameter. Thus we obtain the relative bearing clearance, which is denoted by $\psi = \frac{D-d}{d}$. For the usual proportions ψ averages about 0.001 to 0.002; the designed bearing clearance amounts therefore to 0.001 to 0.002 of the journal diameter.

Further, the actual thickness h_o of the layer of lubricating oil is not expressed as such, but on the contrary in relation to the designed half clearance (R-r). This relative oil film thickness will be noted by $\delta = \frac{h_o}{R\text{-}r}$. For the lowest journal position the lubricating gap is zero, $\delta = 0$: for the concentric position of the journal and bearing $\delta = 1$.

If, further, η signifies the absolute dynamic viscosity of the oil, ω the angular velocity and p_m the mean unit surface loading, then there results from the hydrodynamic lubrication theory the relation

$$\frac{\eta\omega}{p_m\psi^2} = f\left(\delta, \frac{b}{d}\right)$$

It should be noted that in a given position of the journal δ has a wholly definite value and accordingly, for certain width ratios, so has the function $f\left(\delta, \frac{b}{d}\right)$. That is to say, for every position of the journal the expression $\frac{\eta\omega}{p_m\psi^2}$ has a well defined value. Therefrom it follows: the journal position and the thickness of the film of lubricating oil are maintained, in case the unit loading is doubled and an oil of double the viscosity is employed, because then the quotient $\frac{\eta}{p_m}$ will not be changed. Or should the speed be reduced to a half, the position of the journal and the thickness of the oil film remain unchanged if an oil of double the viscosity is used, since the product $\eta\omega$ remains constant.

The correlation between the four factors expresses now the law of similitude, which we can formulate as follows: for geometrically similar positions of the journal in bearings, the expression $\frac{\eta\omega}{p_m\psi^2}$ is equally large, i.e., geometrically similar positions of the journal are characterized by equal values of δ and $\frac{b}{d}$.

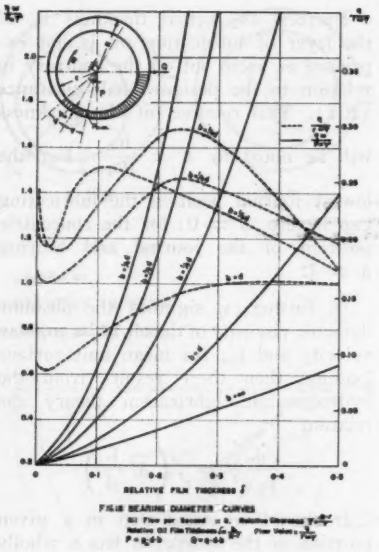
To ascertain the correlation between the relative thickness of the oil film, δ, and the expression $\frac{\eta\omega}{p_m\psi^2}$, and thus to give the value of δ corresponding to the assigned value of $\frac{\eta\omega}{p_m\psi^2}$, is the function of the hydrodynamic theory. In the solution of this problem we are dealing with the solution of the basic equation of the flow of viscous fluids. This partial differential equation becomes solvable only for bearings of infinite width, since then it reduces to a simple differential equation. In cases of bearings of finite width, which interest us as essentially practicable, only approximate solutions are known. In the literature we find, to be sure, approximate formulae for bearing calculations which give, however, only approximate results and which in particular do not take into consideration the influence of finite bearing width satisfactorily. From the nature of the formulae it is clearly evident that they may be useful directly for the calculation of an individual case. However, these do not bring about the understanding of the correlation and dependence of the separate factors, which is indispensable for most practicable bearing dimensioning, for it is unusual in bearing design that we are concerned with one to-be-sought-for factor, but for several participating factors which are conveniently in tune with one another.

As a matter of fact, it is clear that it is hardly possible and makes little sense to force approximate solutions of the basic equation for oil flow into practical applicable formulae which could serve the engineer as a satisfactory basis for proper bearing calculations. Therefore,

a graphic presentation of the solution has been realized, in which the correlation can be seen. The graph (Fig. 18) contains two families of bearing characteristic curves (11). The solid lines show, for five different bearing width ratios, the relation between the relative lubricating oil film thickness as absiscae and the expression $\frac{\eta\omega}{p_m\psi^2}$ as ordinates.

With the help of these curves the bearing calculations proceed very simply. In order to show this it will be explained how we determine the viscosity of the oil necessary, or can balance the surface pressure and the viscosity one against the other. The lubricating oil film thickness h_o, which we wish to obtain, is selected for the appropriate surface quality of the sliding faces. The bearing clearance is selected in relation to the fit, then $\delta = \frac{h_o}{(R\text{-}r)}$ is easily found. From the family of curves the value of $\frac{\eta\omega}{p_m\psi^2}$, corresponding to the δ, is picked off for the selected bearing width ratio. The angular velocity ω is given, the bearing clearance ψ we have selected, and now we can calculate the viscosity η of the oil for the given surface loading p_m or, when the last is not fixed, both the participating factors can be balanced. When other factors are given or are to be found, we proceed in an equally simple way.

A further very essential advance in bearing design was realized when the laws of similitude of the hydrodynamic lubrication were successfully extended to the magnitude of the oil flow and to the heating of the oil of the lubricating film. During each second the oil flow "Q" through the bearing is found as the dimensionless quantity $\frac{q}{r\omega\psi}$, which will be named obviously the "flow factor."



In this, "q" $= \frac{Q}{db}$, the oil flow per second per unit projected bearing area.

In Fig. 18 the graph indicates by dotted lines the magnitude of the "flow factor," with the help of which the required oil flow can be calculated simply. The value of $\frac{q}{r\omega\psi}$ corresponding to δ is read off and "q" calculated therefrom, and ultimately the oil flow per second, qdb = "Q".

In the same way a dimensionless quantity as a measure of the temperature prevailing in the oil film was found. This, the same as the "flow factor," depends in every case only on the relative depth of the lubricating layer, and the bearing width ratio b/d. It can therefore be computed and is not confined to a particular case or to a particular system of measurement.

This quantity we will call θ; it is

$$\theta = \left(\frac{cy}{A} - \frac{w_a}{q}\right)\frac{\vartheta - \vartheta_a}{p_m} - \frac{w_a}{q}\frac{\vartheta_o - \vartheta'}{p_m}.$$

In it "c" signifies the specific heat of the lubricating oil; for mineral oil it has values $c = (0.42 + 0.001\vartheta)$ kcal/kg°C, the density (kg/cm³), A the mechanical equivalent of heat (1/A = 42,700 cm kg/kcal), ϑ the temperature in the lubricating film, ϑ_o the entering temperature of the oil, ϑ' the ambient temperature, w_a the coefficient of heat conduction of the bearing; according to the work of Schiebel (12), the values are:

$w_a = 0.16$ kg/cm sec °C for light construction

$w_a = 0.25$ kg/cm sec °C for a bearing of common design

$w_2 = 0.44$ kg/cm sec °C for a heavily built bearing with large steel masses.

For higher speeds of rotation, the simplified formula suffices:

$$\theta = \frac{cy}{A}\,\frac{\vartheta - \vartheta_o}{p_m}$$

With the help of θ the oil film temperature is easily and simply calculated in cases where the temperature of the entering oil is known, as, for example, if at room temperature.

In Fig. 19 curves for various values of the heating index θ are presented which show its relation with the width ratio b/d and with the relative depth of the layer of lubricating oil δ. From these curves we can observe the following:

(1) The heating of the oil film increases as the film thickness increases. Therefore, δ should be chosen only as large as is necessary to support pure floating friction.

(2) Surely the curves indicate a minimum value for θ at values of $\delta = 0.02$ to 0.03 for the width ratios between 0.4 and 1. Highly loaded bearings run naturally with low values of δ some 0.05 to 0.15; with these values the lowest film temperature is to be expected with

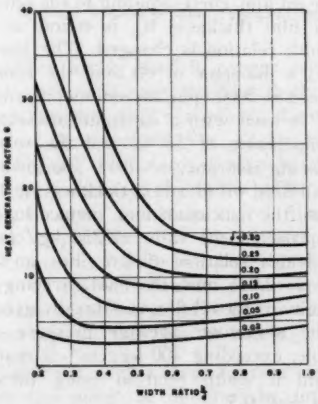


FIG. 19 VARIATION OF HEAT GENERATION FACTOR θ FOR VARIOUS VALUES OF WIDTH RATIO $\frac{b}{d}$ AND FILM THICKNESS δ

bearing width 0.4d to 0.6d. However, the temperature in the film of lubricating oil is determined by the capacity. Factual experiments, which are checked from many sources, have proven that the surface loading is at its greatest at these width ratios.

These conclusions will be an example of how, with the help of the bearing data curves, it is possible to gain knowledge which lays down guide lines for drawing up designs.

GUIDE LINES FOR DESIGN

Width — diameter ratio — We have just now established from the point of view of small heat generation in the oil film that the most favorable width relation for the film thickness corresponding to the loading lies usually between the values of 0.4 and 1. The higher the surface pressure becomes, the smaller the film thickness; accordingly a smaller width relation becomes the most favorable.

Fig. 20 indicates the result of a numerical investigation such as might be accomplished with the help of the bearing data diagram. The pressure in the oil film, corresponding to the selected oil film thickness h_o, is found as the width relation is changed. The journal has a diameter of 60 mm, the rotative speed is 2000 rpm, an automobile engine oil is used with a constant entering oil temperature of 55°C, and the relative bearing clearance is .0015. To form the stabilized oil film at a thickness of 0.002 mm, the maximum unit surface loading approaches the value of 250 kg/cm² at a width relation of 0.6. For a value $h_o = 0.001$ mm, the load carrying capacity of the oil film is naturally greater. The maximum average pressure possible, exceeding 400 kg/cm², is reached with a width relation lying between 0.4 and 0.5.

However, there is yet another argument which speaks against the use of the large width relation, namely: the deformation of the journal and shaft. The mathematical investigation will show that the unavoidable curvature of the journal incident to a bending moment produces a deflection of the journal within the bearing width which, in the case of b/d ratio of 1, amounts now to 0.0001 of the journal diameter. Thus this distortion is reaching an order of magnitude of the film thickness. Accordingly, it would appear that on this basis, and especially in connection with highly loaded bearings, large bearing

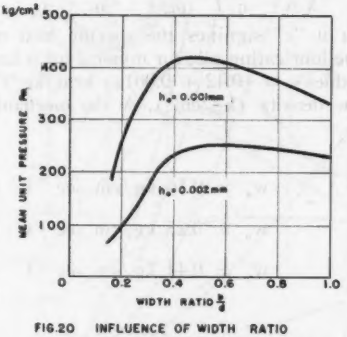


FIG.20 INFLUENCE OF WIDTH RATIO ON UNIT LOADING CAPACITY OF OIL FILM FOR VARIOUS LUBRICATING OIL THICKNESSES h_o

width relations of 1 are not expedient. As a lower limit for usual cases $b = d/3$ is reasonable.

Bearing Clearance—Hitherto, for the want of a systematic approach, clues were rarely found as to the magnitude of suitable bearing clearance. Hence, for identical duties and operation radical differences in clearances were specified. Here, too, the bearing diagrams with their curves make possible methods of obtaining bearing clearances based on investigating the operating conditions and of recommending especially desirable ranges of clearances.

The influence of bearing clearance on operating conditions will be shown with two especially chosen extreme examples: one, a bearing for an automobile engine of 2000 rpm; the other, a bearing for a machine to operate at only 94 rpm.

The first bearing has a diameter of 60 mm, b = 30 mm, thus b/d = 0.5. It is lubricated with a usual automobile engine oil of a viscosity of 15°E corresponding to $\eta = 1.02 \times 10^{-6}$ kg s/cm² at 50°C. In Fig. 21 are presented the oil film thicknesses h_o and the oil film temperature factor ϑ in correlation with the relative bearing clearance ψ for two loadings, 85 kg/cm² and 170 kg/cm². When-

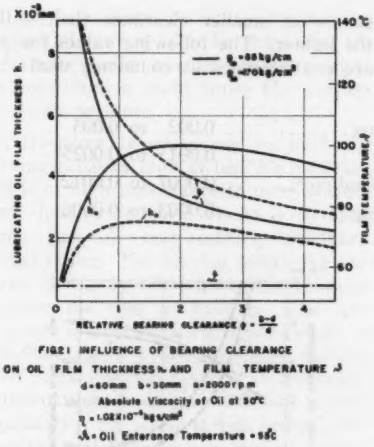

LUBRICATING OIL FILM THICKNESS h
FILM TEMPERATURE ϑ
RELATIVE BEARING CLEARANCE ψ = (D−d)/d
d = 60mm b = 30mm n = 2000 r p m
Absolute Viscosity of Oil at 50°C
Oil Entrance Temperature = 50°C

FIG 21 INFLUENCE OF BEARING CLEARANCE ON OIL FILM THICKNESS h_o AND FILM TEMPERATURE ϑ

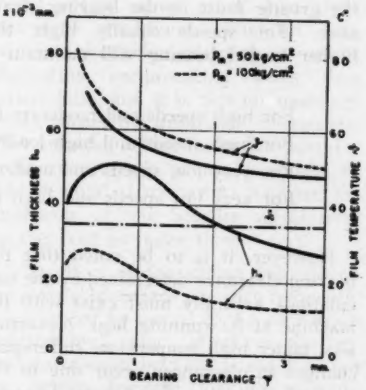


FIG 22 INFLUENCE OF BEARING CLEARANCE ON FILM THICKNESS AND TEMPERATURE

ever the bearing clearance is chosen as less than .001, we recognize that the lubricating film is alarmingly thin and that also the temperature in the film is very high. On the basis of this circumstance we shall choose, in the case of highly loaded bearings, the fits so that the bearing clearance is limited to the range of about .0015 to .0025. For many this is perhaps an unusually large clearance. However, practical experience confirms this to be correct. Because of the smaller heat generation in the oil, it is generally more advantageous if the clearance turns out to be something too large than too small.

Fig. 22 shows the results of the investigation of the second example. However, in the case of the smaller rotative speed of 94 rpm, floating friction even at high loading is certain, once the bearing is correctly proportioned. Inasmuch as with a greater width distortion decreases the capacity of the oil film, b = d = 600 mm is selected as the bearing width. For a surface loading of 50 kg/cm² the thickness of the lubricating film is permitted to reach 0.04 mm, which suffices for floating friction. Even for surface loading of 100 kg/cm², which is higher than in proper accord with such a bearing, the film thickness is greater than 0.02 mm. From the contour of the curves in Fig. 22, the relative bearing clearance will appropriately be kept to between the values .0004 and .001, in order to obtain an oil film as thick as possible. For the diameter of 600 mm, this results in a bearing clearance of 0.3 to 0.6 mm, which seems perhaps uncommonly large but in reality is correct for this type of bearing.

Naturally, such a large bearing clearance can never be obtained by fitting the bearing shells to the journal. It is much better to require that every bearing be bored to the correct diameter ready for installation, to be as smooth and as clean as possible so there will be no necessity for hand fitting. In the case of fitted bearings, the essential wedge-shaped lubricating oil gap cannot form. Such bearings can only be lightly loaded and, even so, constantly produce high running temperatures. For the same reason, adjustable bearing shells should be rejected.

Consequently, for bearing clearance the rule applies—the higher the speed,

the greater must be the bearing clearance. For speeds equally high, the higher loaded bearing will maintain a somewhat smaller clearance than will the lighter. The following values for ψ are recommended as commonly used:

For high speeds and moderate loading	0.002 to 0.003
For high speeds and high loading	0.0015 to 0.0025
For very low speeds and moderate loading	0.0007 to 0.0012
For very low speeds and high loading	0.0003 to 0.0006

However, it is to be noted that the bearing clearance determined by the calculations naturally must exist with the machine at its running heat. Nevertheless, under high temperature differences changes in clearances occur due to the thermal expansion of the parts. In order to provide safety, the clearance will be gauged with the cold machine and with the machine at running heat.

For many purposes the values quoted will be held as too large and quite so, when the duty requires high speeds and light loadings and when the bearings have a special alignment function. Such an example would be the bearings of a steam turbine, of a centrifugal blower, or the spindle bearing assembly on a precision machine tool. When such a bearing fails in service, the reason usually is found to be that in operating at high speed and loaded chiefly by weight forces vibration of the journal occurs within the bearing clearance. This vibration can produce not only erratic running but, even more serious, it may cause destruction of the bearing and even of the machine foundation. Stodola was the first to point out this phenomenon (13). Thorough investigations (14) have disclosed that in such cases of high speeds and light loadings the journal falls into a state of unstable equilibrium and begins to chatter. In such a case exceptional dimensions of bearing clearance must be selected, which cannot be discussed in detail here.

Load Capacity Limits—A detailed discussion of the limits on the load carrying capacity of bearings cannot be

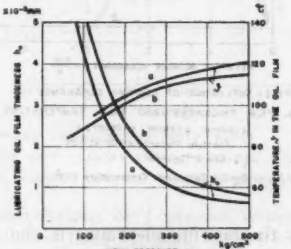


FIG 23 RELATION BETWEEN FILM THICKNESS AND AVERAGE PRESSURE IN THE FILM FOR TWO DIFFERENT OILS

n=2000 r p m b/d=0.5 ψ=0.0015
Oil Entrance Temperature ϑ_e=50°c
a-Automobile Lubricating Oil 15°E at 50°c
(η=1.08×10⁻⁶ kgs/cm²)
b-Aviation Oil 24.5°E at 50°c
(η=1.68×10⁻⁶ kgs/cm²)

offered in this short summarizing presentation, with reference to the various physical influences and correlations. The following brief ideas on this question are advanced as being of possible value as a spotlight.

The lubricating film in itself is considered as being capable of absorbing considerable surface pressure and transmitting it. Fig. 23 shows the correlation between film thickness h_0 and the average film pressure "p_m" in a motor bearing obtainable with two different oils. If a particularly good surface is provided through lapping, grinding and diamond turning, then a very thin film of the lubricant can be permitted. With this a pressure of more than 300 kg/cm² can be absorbed without the occurrence of metallic contact. As a result, the temperature in the oil film increases, but only moderately, as Fig. 23 allows us to see. However, these high pressures

must be withstood also by the bearing materials and it is to be noted that the pressure peaks reached in the oil film range four to eight times the average surface pressure.

However, in discussing the load carrying capacity limit of bearing materials it is of high importance to note whether the loading is static or, as is so often the case in reciprocating machines, alternating. The bearing metals fall into two classes in relation to their behavior under the type of loading. The first group is characterized, accordingly, to include those which are more suitable for static loadings than for reciprocating alternating loadings. To this group, primarily, the white metals belong. For these, if the loading exceeds a certain magnitude which can be endured by the material without difficulties as long as the load is static, a spalling of the material is observed in case of rapidly alternating loads. The material crumbles out, which is characteristically recognized in that destruction does not proceed from the surface but from a spalling at the bond.

In order to clear up these differences with numerical data, the capacity of white metal WM80F should be limited to 120 to 140 kg/cm² under alternating loading and should be utilized only in the most exceptional cases, while however under steady loading it endures 850 kg/cm².

The bearing metals of the second class behave fundamentally quite differently. To this class belong, for example, the high grade lead bronzes and the light metal alloys. These, under alternating loadings, withstand two to three times as much as can be carried under static loads. This difference arises presumably from the inherent behavior of the lubricating oil film under the rapidly alternating loadings. Under these rapidly reciprocating loadings the peak of the pressure has been reached and passed before the journal can follow to attain the attitude of equilibrium that would obtain under other than these transitory conditions. The oil film, therefore, is thicker under these rapidly alternating reciprocating loads than under static, and it is this circumstance which causes the difference in capacity. This essential difference in the behavior of bearing metals primarily should be kept in mind if we are to compare the limitation of one bearing metal over another and to judge them.

The effect of the sliding velocity on the allowable surface pressure frequently is expressed by means of the very much debated "pv" value. However, the factual correlation between p and v is not so simple, for the loading capacity limitations are dependent on three primary factors, which in turn are correlated with other magnitudes. These primary factors are:

1. The hydrodynamic load carrying capacity of the oil film.
2. The tenacity of the bearing metals used, which depends on the structure.
3. The sliding characteristic of the bearing metal with the material of the journal (body friction) and with the lubricating medium.

The systematic plotting of the relationship of the sliding velocity appears in Fig. 24. At first the advantageous influence of the sliding velocity pre-

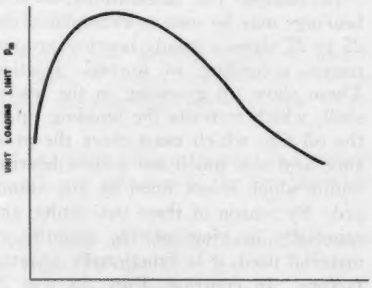


FIG. 24 TYPICAL CURVE
OF STATIC LOAD LIMITS OF A BEARING METAL

dominates, which is brought about by the circumstance that the hydrodynamic capacity of the oil film increases with the sliding velocity, "v." Generally the maximum capacity of the oil film is reached at the approximate velocity of one or more meters per second. However, at still higher velocities a detrimental factor predominates, in that the heat developed per second by friction increases with the velocity and can no longer be dissipated sufficiently. Only in this range does the curve have approximately the character of the hyperbola, and only here can the "pv" factor be justified. However, the majority of bearings operate in a range of smaller rubbing velocities.

In every case the "pv" diagram indicates that the data of an individual value for the allowable surface pressure tells very little about the ability of a bearing to perform well, and little about the possible application of a bearing material. This can only be estimated with some degree of reliability if the limiting load curve of a material is available and, besides, if the performance of the material under alternating loads is known. Presently, such limiting load curves are unfortunately available only for a few materials.

OIL VISCOSITY

Now the viscosity of the lubricating film is conceived to be about as much a factor in design as the allowable surface unit loading or the bearing clearance. Just as the designing engineer must know which fabricating steels are available to him and what strength they possess, he must know which oils are available and what viscosity they will exhibit at the different temperatures.

The viscosity is known to be strongly dependent upon temperature and on a diagram according to Ubbelohde and Walther this relation appears as a straight line. This diagram is recommended, as it suffices to know the viscosity at two temperatures in order to draw this straight line. The viscosity at any temperature can then be read off. We are to understand that by means of this we can obtain the kinematic viscosity "v". For the bearing calculations, however, the absolute viscosity η is required. The two have the simple relation $\eta = v\rho$, where ρ is the density of the oil at the corresponding oil temperature.

EXAMPLES OF DESIGN

In closing, the illustrations of four bearings may be seen as examples. Figs. 25 to 27 show a steady bearing proportioned according to normal practice. These show oil grooving in the loaded shell, which restricts the building up of the oil film which must carry the pressure, and also much too wide a bearing width which is not fixed by any standard. By reason of these two faults, and especially in view of the quantity of material used, it is functionally unsatisfactory. In contrast, Figs. 28 and 29 show a bearing with dimensions fixed according to standards with modern dimensioned shells. Despite the shorter length—or, correctly said, because of it—its carrying capacity is greater. Worthy of note primarily is the substantial lesser quantity of material used, which through comparing the weights is particularly striking. While the old style bearing weighs 15.7 kgs, the weight of the new is only 7.3 kgs: this indicates a saving of over 50%.

The oil grooves of the crankshaft bearing (Figs. 30 to 32) are supposed to lead the oil back to the middle of the bearing, if the direction of rotation is as shown. It may be left undecided

whether this is true or whether the oil still flows chiefly in the direction of the pressure gradient, this from the center outward. In any case, the cutting of the sliding surfaces is lowering the capacity of the oil film to transmit pressure and is therefore detrimental. The fact that bearings with adjustable bearing shells are no longer considered as good practice has been noted previously in the discussion of bearing clearance. Due to the fitting of the bearing shells to the journal, the wedge of lubricating oil in the gap cannot be formed; the hydrodynamic capacity of the oil film is therefore lacking.

In Figs. 33 to 35 a bearing is presented which could not be made wider for lack of space. In spite of the low speed of 6.6 rpm and the relatively high surface loading of 36.3 kg/cm², the controlling factors can be so tuned, one to the other, that even this bearing runs with floating friction. In this case, too, the material saving in the design is of particular importance.

To be sure, with the above treatise only an excursion can be offered in this field of modern sleeve bearings. However, it may convince one that the capacity of these sleeve bearings can be substantially enlarged, especially as used in common machines. This gain is accomplished not through the use of some new, particularly first rate bearing materials but, on the contrary, this will be accomplished generally through utilizing, in a sound approach to the design, a proportioning in accord with the present state of our factual knowledge. Above all, the better ability of a bearing to perform will make itself felt through a substantial decrease in weight. And what is more, the reliability and the durability increase, which circumstance naturally works out to be of equal importance.

Finally, we should consider that the development in the art of building machines will always press toward higher loadings on one side, and, on the other, toward higher rotative speeds. In these developments only those designers will be successful who have acquired a general knowledge of friction and lubricating phenomena, well supported by science and practical experience.

References

1. From a presentation of a paper before an individual chapter VDI im NSBDT.
2. The ringgroove in the bearing shell, lower right, is not lubricating in function, as it serves to further lead the oil to a crankpin bearing. In calculating, each bearing half is understood to be a self-sufficient bearing.
3. Of the two commonly used nomenclatures to note bearing width or length, uniformity favors the former since, in cases of thrust bearings and sliding shoes, it is the custom to denote as width the direction perpendicular to the direction of the sliding velocity.
4. Compare contrarywise, for example, Konstruktive Lagerfragen, 2nd edition, Berlin 1940. On various pages of this work, the influence of the bearing width is shown.
5. E. Heidebroek: Maschinentechnische Anspruche an Schmieröle und Schmierfette. Angew, Chemie, vol. 50 (1937), pages 743-47.
6. By which the shape of the sliding surfaces can be changed to such an extent that the pressure in the oil film will be no longer distributed as shown in the figures.

7. Compare A. Baske: Versuche mit Leichtmetallagers in Prüfmaschinen und Flugmotoren. Automob.-techn for example 42 (1939), pages 335-63.

8. Compare Konstruktive Lagerfragen. 2nd ed., Berlin 1940, pages 5 to 6.

9. Cases during starting periods of the engine should be considered as the exceptions. Then the shaft is lifted by oil pressure from below in order to reduce friction.

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THE CHEMISTRY OF MARINE CORROSION

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The answers to many shipboard maintenance problems are best found by trial and error, but the answer to the corrosion problem can be found in this way only at a high cost in labor, time, and material. A brief study of the chemistry of corrosion will show that rust-preventive practices followed aboard ship are sometimes futile and occasionally harmful. This article will explain the theory of corrosion of metals and show how a knowledge of this theory can be put to practical use.

Corrosion is defined as a chemical change in which a metal combines with a non-metal to form a compound. In general the corrosion product lacks the metallic properties (hardness, tensile strength, elasticity, ductility, etc.) and is therefore undesirable. There are two types of corrosion: direct chemical attack, and galvanic action. The former is exemplified by zinc dissolving in acid. It is rarely met on shipboard because corrosive liquids like concentrated acid are not generally handled; it is important, however, to the chemical industry and others using strong reagents. The second type of corrosion is of interest to the naval officer. In galvanic corrosion a miniature battery is set up in the ship's hull and the adjacent salt water. This battery obeys all the laws of electro-chemistry.

Whenever two electrically connected metallic objects of different electrochemical activity are immersed in a current carrying solution (called an electrolyte), the atoms of the more active metal become positive ions by giving up one or more electrons. The positive ions then combine with negative ions from the electrolyte to form a compound—the corrosion product. The electrons given up flow through the electrical connection to the less active metal and there are picked up by a positive ion in the electrolyte. The less active metal is not affected while a part of the more active metal has turned to rust.

Almost any liquid with the exception of the most carefully distilled water will serve as an electrolyte. There are two

ways in which the two metallic objects of different electrochemical activity may be present. If there are any mechanical strains in a metal object the activity of the strained point is greater than that of the rest of the object. For this reason more rust occurs at the head and point of a nail than at other parts, and rusting begins in the vicinity of rivet holes, sharp bends, and other regions that have been cold-worked. Electrical contact exists, of course, between all points on a single metal piece. Secondly, different metals have inherently different activities, largely determined by the structure of the atoms. Although certain factors, including cold work and concentration of the electrolyte, may change the order, the usual relative activity of the common metals is as follows:

magnesium (most active)
aluminum
zinc
chromium
iron
cadmium
nickel
tin
lead
(hydrogen ion)
copper
silver
platinum
gold (least active)

In the case of two dissimilar metals there must be an electrical contact between them other than the electrolyte. They may be in physical contact or may be connected with a wire.

A corrosive reaction which often occurs aboard ship can be represented by the following chemical equations and by the accompanying Figure.

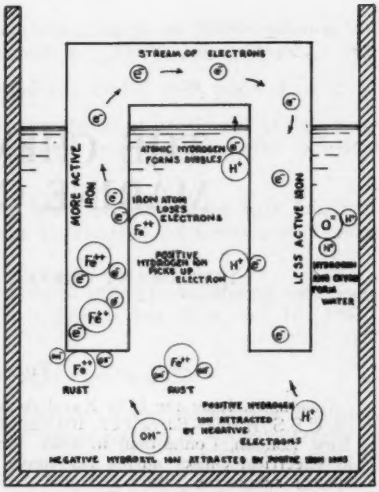
$$Fe \rightarrow Fe^{++} + 2e^-$$

$$H_2O \rightarrow H^+ + (OH)^-$$

$$Fe^{++} + 2(OH)^- \rightarrow Fe(OH)_2 \downarrow$$

$$2H^+ + 2e^- \rightarrow H_2 \uparrow$$

$$2H_2 + O_2 \rightarrow H_2O$$


STREAM OF ELECTRONS
ATOMIC HYDROGEN FORMS BUBBLES
MORE ACTIVE IRON
IRON ATOM LOSES ELECTRONS
POSITIVE HYDROGEN ION PICKS UP ELECTRON
LESS ACTIVE IRON
RUST
RUST
POSITIVE HYDROGEN ION ATTRACTED BY NEGATIVE ELECTRONS
NEGATIVE HYDROXYL ION ATTRACTED BY POSITIVE IRON IONS

GALVANIC ACTION

Here iron (in the steel hull) is the metal being corroded. Since iron is more active than the hydrogen ion H^+ of the water (as seen in the table, it gives up two electrons and combines with the hydroxyl ion $(OH)^-$ of the water. (Copper, on the other hand, is less active than the hydrogen ion and hence could not displace it from the water.) The two electrons given up travel through the metal until they meet two hydrogen ions at the surface. They take up the electrons to become neutral hydrogen, which combines with oxygen dissolved in the water to form more water. Unless this hydrogen is removed it tends to form a layer of bubbles over the iron which prevents contact between iron and electrolyte and stops the reaction. The presence of oxygen, which removes this film, accelerates corrosion as evidenced by the pitting formerly found in the unpainted fuel ballast tanks on submarines. After the fuel was expended these tanks were used for main ballast. The constant blowing and flooding kept the oxygen content high, and severe rusting occurred until the de-

velopment of a plastic tank coating not affected by fuel oil or salt water. Since iron, water, and dissolved oxygen are always present in the vicinity of a ship's hull, this reaction occurs continuously. Prevention of this reaction is one purpose of deaeration of boiler feed water.

A similar reaction takes place whenever a "couple," or combination of two dissimilar metals in electrical contact, is immersed in an electrolyte. In the case of iron (less active) and zinc (more active) the following reactions occur:

$$Zn \rightarrow Zn^{++} + Ze^{-}$$
$$H_2O \rightarrow H^{+} + (OH)^{-}$$
$$Zn^{++} + 2(OH)^{-} \rightarrow Zn(OH)_2 \downarrow$$
$$2H^{+} + Ze^{-} \rightarrow H_2 \uparrow$$

The iron is not affected, although a stream of electrons flows through the iron into the ions in the solution.

Galvanic action causes three types of corrosion: general corrosion, pitting, and intercrystalline attack. As its name implies, general corrosion is a more or less uniform reaction which causes large areas to be rusted to about the same depth. It is usually the least dangerous form of corrosion. Pitting is localized, severe corrosion which may extend for a considerable depth into the metal. In applications where strength and tightness of the member are important, such as a submarine pressure hull or a high pressure air flask, this type of attack is especially dangerous. Not only does it badly weaken the metal, but it is also hard to detect. Intercrystalline attack is a special type of corrosion which occurs in brass, stainless steel, and a few other alloys. It is sometimes called season cracking or caustic embrittlement. In this case corrosion proceeds along the boundaries of the minute crystals of which all metals are composed. In extreme cases the metal crumbles in the same fashion as a brick chimney with the mortar removed, the bricks corresponding to the crystals and the mortar to the intercrystalline matter which holds the crystals of a metal together. Such failure is often incorrectly explained by saying that the metal has "crystallized." Actually all metals are composed of crystals so minute that they are ordinarily invisible. When a metal part is fractured, its crystalline form is more evident along the broken surface, and this appearance has created the false impression that crystallization causes the failure.

Because of the zinc-iron reaction described above, zinc protectors are mounted on the hull and in the piping of ships to minimize corrosion in the vicinity of electrochemical couples. At the stern of a ship, for example, there exist couples between the steel hull and the bronze propellers. Severe corrosion might be expected here; the less active bronze would not be damaged while the nearby iron would rust badly. Placing a zinc plate nearby introduces another couple which causes the zinc to be corroded, since it is more active than the iron, while the iron and bronze are not damaged.

Zinc plates are occasionally secured to the hulls of ships by methods which render the zincs almost useless. In one instance the inside of the zincs was given a thick coat of paint just before the plates were bolted into place. Fortunately the bolts were not painted and were able to supply the electrical connection to the hull necessary for the zincs to supply any protection at all. At subsequent drydockings, the ship's force was unable to explain why bolts and occasionally entire plates had disappeared. The reason is obvious; the bolts corroded away because they were in better electrical contact with the iron hull than were the plates. When all its bolts rusted out, the plate dropped off without affording the protection for which it was installed. The moral is to ensure good electrical contact between the hull and the zinc by removing all paint and rust on the adjacent surfaces.

A common error of painters is to paint over the zincs when they paint the hull. When cautioned, they will often coat the zincs with grease. The grease protects the zincs from paint, but unless removed it will also protect them from salt water after the ship is undocked, and thus render the protector useless. The moral here is to ensure contact between the zinc and the sea water.

When the first high speed Guppy submarines were drydocked it was found that the zincs were more corroded than those of the slower fleet-type boats. This accelerated corrosion was caused by removal of the layer of hydrogen bubbles formed where hydrogen ion is changed to atomic hydrogen on the surface of the iron. Higher speed swept away the bubbles and caused the reaction to proceed more rapidly. In general, then, higher speeds mean more corrosion.

Aluminum on the underwater body of a ship corrodes rapidly, while aluminum immersed in still water forms a thin coat of corrosion product and then stops corroding. Since aluminum stands high on the electrochemical series it corrodes rapidly, but the oxide produced forms a uniform watertight adhesive film which protects the metal underneath from further corrosion. If this film is removed by erosion of moving sea water, however, the attack continues until the metal disappears. Iron rust forms in flakes instead of a uniform film and moreover is porous and absorbs salt water. Instead of protecting the metal beneath it accelerates corrosion in that area. The metal underneath the rust is in contact with an ample supply of electrolyte (sea water) but is shielded from absorbed oxygen while the exposed iron has a greater oxygen supply. Oddly enough, this oxygen differential causes corrosion to proceed more rapidly in the vicinity of the smaller oxygen supply. The rust spot eats into the iron and causes the dangerous condition known as pitting.

A film of corrosion product which covers the entire metal surface and is impervious to sea water protects the metal; a non-uniform or porous coating damages the metal. A protective coating on iron and steel can be chemically produced by immersing the part in a hot solution of phosphoric acid, iron phosphate, and sometimes manganous oxide. This principle is used in Parkerizing torpedo afterbodies and battery compartments. Compounds like Tectyl form a semi-chemically combined coating of the same general type.

Where couples exist, the relative area of the different metals is important. In a pipe line consisting of copper tubing with iron elbows and tees, the more active iron fittings rust first. The small total surface area of the iron fittings causes the rust to collect there and clog the pipeline. If a couple must be tolerated, iron piping and copper fittings would be preferable. The iron would still corrode, but the rust would be distributed over the entire surface area of the piping and would not clog the pipe so rapidly.

Numerous engineering casualties demonstrate that couples are undesirable. On a certain submarine the flapper valve in the bowl of a water closet developed leaks. Both valve and disc were manufactured of bronze and quickly became scored. Although the disc was readily removable for repair, the valve seat was not, and lapping the seat in place became a difficult and unpleasant task. Thinking to stop wear of the valve seat, an auxiliaryman replaced the bronze fitting with a harder seat of stainless steel. The valve disc, he thought, would then take all the wear and the chore of keeping the valve tight would be simplified. His plan failed, however, because he introduced a couple. Although he stopped erosion and scratching of the valve seat, galvanic corrosion set in and the valve leaked worse than ever. It was finally

necessary to replace the original bronze fitting and accept abrasion rather than corrosion.

A more serious casualty occurred on another submarine in which an iron pipe plug had been installed in a bronze pipeline carrying circulating water to the ice machine. Undetected by the operating personnel, the iron was corroding from the outboard end with the result that sea pressure ultimately blew the plug out of the pipe and the pump room began to flood. The ship was on the surface and no serious damage resulted, but the same casualty at deep submergence would have been dangerous.

Concentration and temperature of the electrolyte affect the rate of corrosion. Current flow is a function of the number of ions in solution; consequently salt water is a better conductor and causes worse corrosion than fresh or distilled water. Since most brazing and soldering fluxes also ionize in water, they should be removed from metal surfaces after the joint is cool. High temperature in general accelerates corrosion as it does other chemical reactions. There is one exception where high temperature inhibits corrosion. The solubility of most salts occurring in "hard" water varies inversely with temperature. Tubing carrying hot hard water often becomes coated with a scale which, while decreasing the cross section and the rate of heat transfer of the pipe, protects the metal from corrosion.

Several alloys have been developed which combine good mechanical properties with high resistance to corrosion. The brasses (copper plus zinc) meet these requirements and are widely used aboard ship. Alloys of this type, where the component metals are mixed on a microscopic scale, do not form couples even though two dissimilar metals are exposed to an electrolyte. It is only when the two metals are present in particles of macroscopic size that galvanic corrosion occurs. Monel, aluminum bronze, and beryllium copper are other non-ferrous alloys with excellent mechanical properties. The latter is even suitable for edges of cutting tools. Used by the Inca Indians of Peru at the time of the Spanish Conquest for spear points and knives, copper-beryllium alloys were subsequently forgotten until their rediscovery in 1926. Among the ferrous alloys the stainless steels, often known as "C.R.S.," are most important. Since the term corrosion-resistant is applied to steels which exhibit this property in varying degrees, it is important to choose a steel which will stand the service intended. The cheaper grades, with rather high nickel content, will resist corrosion from atmospheric moisture but are unsuitable for use below the water line. All alloys mentioned, and most other corrosion-resistant alloys, are non-magnetic. A simple test for corrosion-resistant metals is to apply a small permanent magnet. If the magnet is attracted to the metal it is likely to corrode and vice versa. A more elaborate test with nitric acid indicates qualitatively the composition of the alloy tested. Table I below lists a number of non-corroding metals with the approximate composition and some of the important properties.

Plating with corrosion-resistant metals has certain advantages but may boomerang and cause severe rusting. Galvanized, or zinc plated iron is always safe. Since zinc is above iron on the electrochemical series, the zinc will always corrode before the iron even if the plating is pierced and the iron is exposed to sea water. Tin plating over iron, on the other hand, prevents corrosion only if the tin plate remains unbroken. If even a small scratch penetrates the tin and exposes the iron, galvanic action will begin. Iron stands higher on the series than does tin, and consequently the iron corrodes at a rapid rate while the "protective" tin is undamaged. Corrosion will continue under the scratch and the iron will tend to rust through. In choosing a plating,

TABLE I

Alloy	Composition Percent		Tensile Strength psi	Brinell Hardness	Remarks
Brasses					
Naval Brass	copper	60	70,000	90	Soft, workable.
	zinc	39			
	tin	1			
Admiralty metal	copper	71	90,000	90	
	zinc	28			
	tin	1			
Aluminum bronze	copper	94	100,000	120	Harder, wear resistant.
	aluminum	5			
	iron	1			
Beryllium copper	copper	98	175,000	350	Hardest non-ferrous alloy—similar to tool steel. Expensive. Non-sparking.
	beryllium	2			
"K" Monel	nickel	63	125,000	250	Similar to mild steel. Machines and welds well.
	copper	30			
	aluminum	4			
	iron	3			
Stainless steel	iron	74	80,000	140	Ductile, non-magnetic.
	chromium	18			
	nickel	8			

then, it is preferable to select a metal which not only resists corrosion but also is more active than the metal to be protected. Cadmium, nickel, and tin are less desirable than zinc and chromium. In practice steel is plated with copper, then nickel, then chromium to combine the tough, impermeable skins of copper and nickel with the hard, anodic chromium layer to give so-called "chrome plate."

There are several other methods of coating the surface of one metal with another. Molten metal can be sprayed on the surface in a process often used to protect welds in galvanized iron. Such a coating must be relatively thin, is rather weak, and being porous must be made of a more active metal than the part being coated. Ihrigizing and chromizing produce coats of silicon and chromium, respectively on steel surfaces by impregnating the surfaces at high temperatures with these metals. In "cladding," a plate of corrosion-resistant metal is joined to a plate of steel by rolling the plates together with high pressure at elevated temperature. The "Pluramelt" process can coat a steel slab of several hundred tons with a two-inch layer of stainless steel. By an elaborate apparatus the stainless steel is cast directly on the coat of the slab to form a perfect union of the two alloys.

In summary, one means of preventing corrosion of a metal is to change its environment by eliminating oxygen or by covering it with paint, metal, or other inert coatings. Another preventive is to use corrosion-resistant alloys. Couples should be avoided and zinc protectors installed where they exist. Finally, if a certain amount of corrosion must be accepted, proper design can minimize the damage.

PARALLEL RANGE MIRROR
A New Navigational Instrument for Determining Compass Errors by Parallel Ranges

JOHN W. SAWYER

THE AUTHOR

JOHN W. SAWYER received his B.S. degree in E.E. from N. C. State in 1933. He served a number of years in the merchant marine. In 1942 he entered the Navy. Since release to inactive duty he has been active in the main propulsion reduction gear development program of the Bureau of Ships.

INTRODUCTION

Instruments with known errors can be used with confidence by personnel in the mechanical laboratory and on board ship. However, to use an instrument without having knowledge of its error and applying the proper correction factor is inviting unnecessary problems. In the case of navigational instruments, on both merchant and naval vessels, it may result in serious trouble, such as grounding or collision. In order for a navigator to rely on a ship's compasses, gyro and magnetic, it is essential that the errors of these instruments be known at all times insofar as practicable.

A number of methods have been devised for determining gyro error. They vary as to time required, conditions of visibility, equipment, and personnel. These methods include: bearing on a distant object (1) from a known position; bearing on a range (2) where both range objects are on the same side of the vessel; bearing on a *center range* (3), which is defined as a range comprised of two known objects, one on either side of the observer; and by celestial sights. There are many obvious advantages and limitations of each method which need not be discussed in this paper.

An instrument, the *parallel range mirror*, has been developed, patent pending, that permits compass errors to be determined by sights on a horizontal line, of known direction, such as the edge of a dock structure, a bridge, or ledge of a building. The horizontal line must, of course, be in a plane that is well above or below the horizontal plane that includes the sight line of the instrument. This instrument is particularly well suited for checking compass errors while alongside a dock, or passing under or near a bridge. The unique features of the instrument, principle of operation, and examples illustrating its use will be described in the following paragraphs.

PARALLEL RANGE MIRROR

DESCRIPTION

This instrument, constructed of nonmagnetic materials, consists of the following basic parts:

Adapter plate
Graduated dial
Bubble level
Parallel range mirror with base and pointers

The adapter plate, with three supporting contact fingers, is shown in Figure 1. This part is designed to permit placement of the instrument on a bearing or azimuth circle similar to the center range mirror (3) shown in Figure 2. A stud for locating the graduated dial and parallel sighting mirror is in the center of the plate. Adjusting screws are located in the end of each supporting finger to permit the instrument to be centered over a gyro repeater or magnetic compass even though the inner diameter of the bearing circle may not be concentric with the card of the compass. The three supporting fingers cover a very small portion of the compass card and do not interfere with an observer's view while reading the vessel's heading. The plate is designed to permit the assembled device to remain in place on the bearing circle while bearings are being taken on fixed objects or other vessels.

The graduated dial, marked in one degree steps in a clockwise direction from 0-360 degrees, is of a transparent plastic material. The dial has a centrally located hole that permits it to be placed over the center stud of the adapter plate. The diameter is such that the lubber's line on the gyro repeater or magnetic compass will extend to a position near the outer edge of the dial. This permits an observer to read the vessel's heading as indicated on the graduated dial by the lubber's line. Since the dial is transparent the compass heading of the vessel can be observed without removing the parallel sighting mirror.

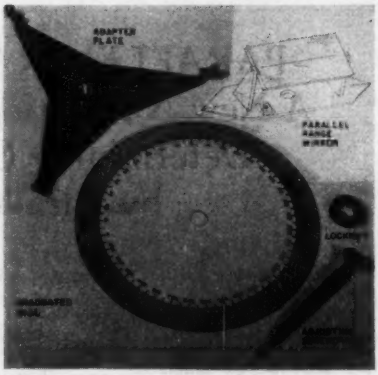


FIGURE 1—The parallel range mirror with components necessary for mounting on a bearing or azimuth circle.

The bubble level, see part 10 of Figure 3, is of the type that will permit the adapter plate with the sighting mirror and dial to be placed in a horizontal plane. It is noted that this is a vertical level and is different from that used on bearing and azimuth circles.

The parallel range mirror is shown in Figure 3. The mirror 1 is pivoted at

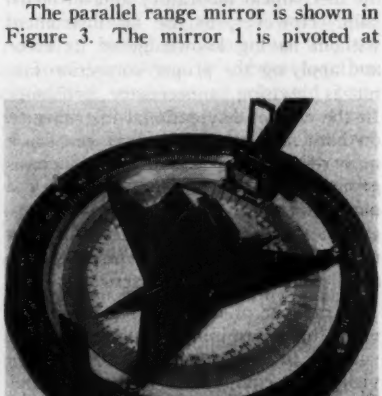


FIGURE 2—The parallel range mirror is mounted on a bearing circle in the same manner as the center range mirror shown above.

points 2 near the top of the supporting members 3 which are fixed on the pointer base 4. It may be tilted in the directions indicated by the arrows. The pointers 5 are of such length that they extend to the degree markings on the graduated dial. The reference line 6, if extended, would pass through the hole 7 in the base. The pivot points are located in the support members so that a line, connecting the pivots, is parallel to line 6. The mirror is constructed so edges 8 are parallel to the index line 9 which is etched on the reflecting surface of the mirror. This line may be treated with luminescent material to facilitate use of lighted ranges under night operations. The construction is such that lines 6, 8 and 9 are parallel at all times regardless of the position of the mirror.

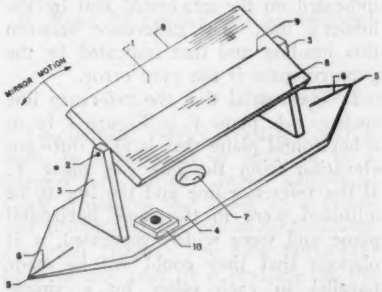


FIGURE 3—Arrangement of the parallel range mirror, supporting mounts and pointers.

PRINCIPLE OF OPERATION

The procedure for determining the compass error by use of azimuth or bearing circles in sighting on various arrangements of known objects was mentioned earlier in this paper. The basic principle of each method employing ranges is the adjustment of a vertical plane, such as one passing between the eye piece and the sight wire of a bearing circle, until it is parallel to a vertical plane connecting two known objects comprising a range. When the two planes just mentioned are parallel the line of sight is in a known direction, namely that of the range. The observer then notes the difference between the true bearing of the range and the bearing by gyro. This difference is the gyro error.

The *parallel range mirror* operates on the principle of adjusting a line in one horizontal plane until that line is parallel with a line of known direction in a second horizontal plane that is at a *different* elevation. This is illustrated in Figure 4. Planes E and F are horizontal. An observer at position "W" can parallel line c-d in plane F to line a-b in plane E by moving c-d until it is superimposed on a-b. Then c-d has the same bearing as a-b.

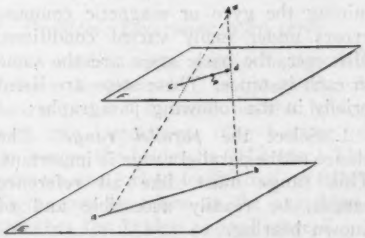


FIGURE 4—Method of paralleling two lines in horizontal planes of different elevations.

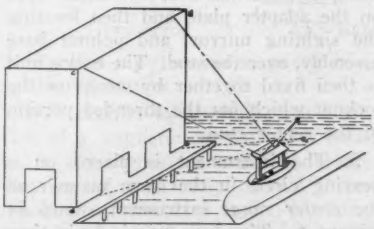


FIGURE 5—Sights taken on dock structure with parallel range mirror.

The principle will be readily understood by referring to Figure 5. The line through the pointers of the instrument may be paralleled to the lines a-b or c-d on the building. The observer

accomplishes this by leveling the instrument and then rotating the mirror on its pivots and turning the bearing circle supporting the unit until the line c-d is superimposed on the sighting line of the mirror. With the eye at e he will see the line just mentioned. He may move to position e_2 and sight across the upper edge of the mirror and parallel this with line a-b of the dock structure. When either of these conditions is reached a line through the pointers is in the same direction as the building lines a-b and c-d. If a-b bears 082° then the pointers will also bear 082°.

Now if the graduated dial is fixed to the pointers so the 082° mark on the dial is parallel to the pointers, then the 0° point on the dial is true north. The true heading of the vessel would be as indicated on the graduated dial by the lubber's line. The difference between this heading and that indicated by the gyro compass is the gyro error.

It is essential that the reference line such as a-b, plane E in Figure 4, be in a horizontal plane that is at a different elevation from the line c-d, plane F. If the reference line and the line to be adjusted were in the same horizontal plane and were widely separated, it is obvious that they could not be made parallel to each other by a simple method.

METHOD OF OPERATION

This instrument is suitable for determining the gyro or magnetic compass errors under many varied conditions. However, the basic steps are the same in each instance. These steps are listed briefly in the following paragraphs:

1. Select the *parallel range*. The choice of the parallel range is important. This range must, like all reference ranges, be readily accessible and of known bearing.

2. Set up instrument. The instrument is assembled for operation by placing the graduated dial over the center stud on the adapter plate and then locating the sighting mirror, and pointer base assembly, over the stud. The entire unit is then fixed together by means of the locknut which fits the threaded portion of the center stud.

3. The instrument is placed on a bearing circle in the same manner as the *center range* instrument shown in Figure 6. The bearing circle is then placed on a gyro repeater.

4. The pointers are locked, by the locknut, to the graduated dial at the true bearing of the selected *parallel range*.

5. The instrument is then leveled. The bearing circle is rotated to a position that brings the *parallel range* into view on the mirror. The mirror is adjusted as necessary. The bearing circle is then turned to a position so that the

FIGURE 6—The parallel range mirror would be mounted on a gyro repeater similar to the center range mirror shown.

index line of the instrument 9 of Figure 5 is parallel to the reflected image of the range c-d.

6. Read the vessel's head as indicated by the lubber's line on the gyro card and the head as indicated by the lubber's line on the graduated dial. The difference in these headings is the gyro error.

SUITABLE PARALLEL RANGES

The parallel ranges available near a dock, in a shipyard or adjacent to a channel are usually quite numerous. Dock structures that block in a vessel and prohibit sights on distant objects are ideal ranges. The long horizontal lines of any part of a building in the vicinity of the dock are satisfactory. Railroad tracks along the dock permit accurate and quick sights. Bridges that cross over a channel are excellent ranges and are to be found in many harbors. Night sights can be taken on illuminated structures. Two colored lights, suitably spaced, can be placed on straight horizontal dock structures to form a *parallel range* for night sights.

ILLUSTRATIONS OF USE

A number of illustrative examples will be given in the following paragraphs. These will adequately set forth the suitability of this instrument under many conditions.

1. Problem: Determine gyro error

The navigator of a vessel alongside a dock desires to check the gyro error. The dock structure blocks view of distant objects that show on his hydrographic charts. Fog prevents use of the sun. The error is quickly and accurately determined by the parallel range mirror and the dock building. The bearing of an edge of the building, say c-d as in Figure 5 is determined from the chart as 127° true. The pointers are locked to 127° on the graduated dial, details of assembling the instrument were described in previous paragraphs. Place the instrument on the gyro repeater, level and adjust until the range c-d is parallel to the mirror index line. Take readings of heading by gyro and graduated dial.

Head by gryo	138°
Head by graduated dial	136°
Gyro error	2° West

2. Problem: Determine gyro error

A vessel is to pass under a bridge that crosses the channel. The gyro error is determined by sighting on the bridge with this instrument. The bearing of the range, the bridge, is 011° true. The pointers are locked to the dial on 011°. Sights can be taken as the vessel approaches the bridge as in Figure 7.

Head by gyro	342°
Head by graduated dial	344.5°
Gyro error	2.5° East

3. Problem: Determine magnetic compass deviation

It is desired to determine the deviation of a magnetic compass of a vessel as it approaches an overhead bridge. Bearing of the bridge is 031° true. Variation in the area is 5° W. Magnetic bearing of the range is 036°. The pointers are locked to the dial on 036°. Sights are taken at some distance from the bridge.

Head by magnetic compass	357°
Head by graduated dial	001°
Deviation	4° East

PARALLEL RANGE MIRROR

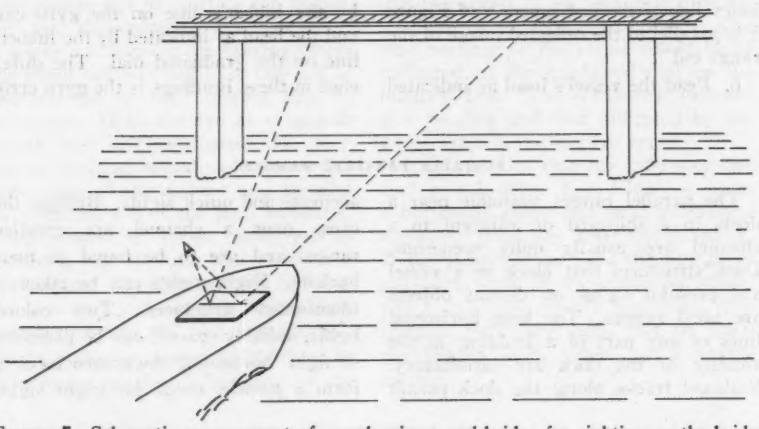


FIGURE 7—Schematic arrangement of vessel, mirror and bridge for sighting on the bridge as a parallel range.

4. Problem: Determine gyro error

A gyro on board a vessel failed just prior to sailing. Yard personnel reported on board and repaired the trouble. Visibility was poor. Neither distant ranges or stars were visible. An edge of the lighted dock structure was used as a parallel range and the gyro error was easily measured.

Head by gyro	138°
Head by instrument dial	136°
Gyro error	2° West

CONCLUSIONS

It is apparent that this instrument provides a very easy and accurate method of checking compass errors while a vessel is at a dock or in a harbor. Its use will permit compass checks in times of limited visibility or in the absence of the usual type range.

Dockside work on gyro compasses by yard personnel may be conducted and gyro errors easily determined by sighting on a parallel range.

This instrument has a number of major advantages that make its use most essential under conditions of poor visibility where *parallel ranges* are available. These advantages include: ease of operation, rugged construction, accurate results and a greatly increased field of operation.

ACKNOWLEDGMENTS

The author wishes to express his appreciation to the Department of Defense for permission to publish this paper and to personnel in the Bureau of Ships and the U. S. Naval Academy for valuable comments and suggestions regarding the preparation and arrangement of the material presented.

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DISCUSSION

COMMANDER EDWIN A. BEITO, USNR

The opinions or assertions here expressed are the private ones of the writer, and not to be construed as official or reflecting the view of the U. S. Naval Academy or the naval service at large.

I was interested in reading the above article by John W. Sawyer and also his article which appeared in this publication, 1 February, 1950, entitled "A Navigational Instrument." The articles explain two recently constructed instruments by which gyro and magnetic compass errors may be determined, especially when the established methods are not possible or not available. The two modes of procedure are referred to as the "center range" and "parallel range" methods.

In both articles Mr. Sawyer vividly describes the instruments and ably illustrates their possible use in obtaining compass errors. His theoretical discussion can well be understood and it would appear that all his claims for the instruments are substantially correct. However, it would strengthen the author's theses immeasurably if he would substantiate his claims by referring to actual results when the instruments are used aboard ship. It would seem well worth while to make a comprehensive study of the practical uses of the instruments to determine their value as standard equipment on ships.

BORON STEELS—A NEW ERA IN ALLOY METALLURGY

ACKNOWLEDGMENT

This article is an excerpt from a series of articles published in the July 5, 12, 19, and 26, and August 9, 1951, issues of "The Iron Age," by D. I. BROWN, Technical Editor of "The Iron Age." It discusses the background of boron steels, with their advantages and limitations. Reprints of the complete series are available from "The Iron Age," 100 East 42nd St., New York 17, N. Y., at a nominal charge.

Critical shortages of chromium, nickel, molybdenum; columbium, tungsten and other alloys mean that boron steels must carry more of the impact of war on both our civilian and defense production.

Fortunately domestic supplies of borax are plentiful. Even more significant is the fact that the new steels were developed, melted, fabricated into parts and tested in less than a third the time it took us to develop the National Emergency Steels of War II. The boron steel alternates for higher alloyed grades are based on equivalent hardenability. In some cases heat treating and carburizing practices must be modified to get optimum properties but on the whole most of these steels behave like the ones they replace.

Boron steels are not new. It is estimated that about 2 million tons have been used since the first commercial heat for actual application was melted in the late 1930's. The original idea of using small amounts of boron to increase hardenability was conceived in the mid-twenties. (1) Today this element is a full-fledged alloying agent. It is much more potent hardenability-wise than the hard-to-get nickel, chromium and molybdenum, long used in alloy steels.

Manganese is also a hardener and boron can be used to replace part of the manganese in the standard high manganese grades. However, it is believed that 0.60 Mn minimum will be necessary in all boron steels, so that the biggest saving in strategic alloys through the use of boron will concern chromium, nickel and molybdenum.

Some of the early claims made for the effect of boron in steel didn't help in its acceptance. Like many new developments, certain conclusions about boron were made prematurely. Thus boron has had all the extra hurdles encountered by child prodigies shoved in its path. It isn't good for everything, it's not a panacea, it must be used judiciously; but properly employed, *boron is today the only bright part of an otherwise very dark and difficult alloy picture.*

A little bit of boron goes a long way. In fact, boron will replace several hundred times its own weight compared to other alloys such as chromium and molybdenum on a hardenability basis. For instance, at the 0.40 C level 0.002 pct B is equivalent to 0.30 Mn or 0.35 Mo or 0.50 Cr or 2.00 Ni. (2) With less than half the required nickel and molybdenum available for the coming limited defense effort requirements, more

alloy steel parts must of necessity be shifted to leaner alloys, boron treated; or just plain carbon boron steel, if there is to be enough heat-treatable steel to go around.

Boron is not as versatile as the other alloys which, in addition to being steel hardeners, promote corrosion resistance, wear resistance or better high-temperature properties. All boron does is to increase the depth to which a steel can be hardened. This characteristic—called hardenability—is achieved by heat treating, usually by quench and draw methods. Boron does not change the actual hardness itself, as hardness is a function of carbon content.

The boron steels of today do have some advantages over former standard alloys. In addition to promoting alloy conservation, boron steels machine better, they have better hot and cold working characteristics, and shorter annealing cycles are possible in most boron analyses.

The NE steels forced on us of necessity in War II taught the American metallurgists plenty about hardenability and alloy conservation. Six of the NE steels stuck, were adopted as standard grades and used in large tonnages right up to the present; although heavy production was confined to only three of these grades, the 8600, 8700 and 9300 series.

Today we can't even afford the NE alloys of War II lean as they are in alloy compared to the former grades. All the 1951 metallurgist has available to make the constructional alloys for gears, axles, pinions, bolts, other heat treated applications and possibly some types of bearings is the residual alloy content obtained from scrap, a little manganese and chromium—*and boron.*

The result has been that since January 1951 62 new alloy grades in 7 types, 4 boron and 3 non-boron steels have been developed which, through the ingenuity of our metallurgical engineers, may see us through. Many of these steels are triple alloys in which the chromium, nickel and molybdenum contents have been pared to the bone. They are being turned out in production heats. and are in use. About 5 new grades will soon be added, 2 of the boron type and 3 more lean alloy, non-boron type. Shops which never heard of boron steels or resisted their use will have to take it or else.

Boron has proved very effective in both carburizing and medium carbon grades. The high nickel carburizing types, 3300, 2500, 4800 and 4600, long used in automotive gears and pinions and shafts, are being replaced by 43B17, 94B17, 94B20, 81B20, 80B20 and 50B20.

BORON SPRING STEELS ARE IN USE

It is generally held that the effect of boron on increasing hardenability decreases as the carbon increases, see Fig. 1. (3) A multiplying factor, Fig. 2, for the effect of boron permits calculated hardenability for any given carbon level. (3) This formula proved very accurate and helpful in some of the work in developing the new boron steels.

It is estimated in some quarters that about 2 million tons of boron steels, excluding armor, may be produced this year. By the end of the year 70 percent to 80 percent of construction alloy production based on 1950 tonnages may be boron grades. Next year it is expected that the boron steel tonnage will greatly increase. Tonnages as high as 20 million tons have been mentioned.

The steel mills have worked out melting procedures for boron on many boron additive agents. All these agents are added in the ladle and pounds per ton added vary from shop to shop and from one steel type to another. However, the usual weight of boron ferroalloy added per ton to reach the minimum boron content required is well established. Usually 50 percent more boron ferroalloy per ton is required in electric furnace heats than for the same steel made in the openhearth.

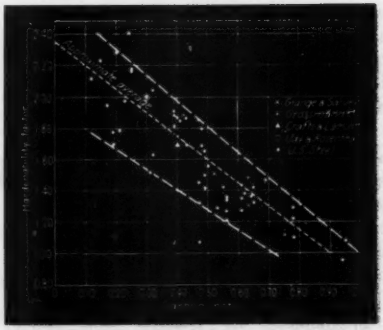

FIG. 1—Effect of carbon content on the hardenability factor of boron. This is based on ideal critical diameters derived from 50 pct martensite critical hardness.[3]

MANY FERROALLOYS CAN BE USED

Although some steel producers and users have certain preferences about the type of boron ferroalloy, all of these agents, when properly added, accomplish the major goal of increased hardenability. (4, 5) Usually the users leave this choice up to the mills. Lately some users have specified boron steels, 94BV17 or 43BV14, which requires the mill to use a vanadium-bearing ferroalloy such as Grainal No. 1.

The proposal that boron be used to replace some of the manganese content of alloy steels has no place in today's picture. If this were done, there would not be enough boron to go around. The ferroalloy producers may be pushed to supply the quantities of boron needed to replace nickel, chromium and molybdenum, but it appears they will be able to meet all boron requirements to make the present types of boron steels. The supply of boron is unlimited. We have deserts full of borax but the electric furnace facilities to make the boron ferroalloys are limited. Electric power shortages are really the only bottleneck which might limit the production of the boron ferroalloys.

Before citing the wartime, postwar and cold war applications and pertinent properties of boron steels, some of the limitations or facts of life on this alloying agent should be understood. While it is firmly believed that the use of boron steels offers the greatest method at hand of alloy conservation, some parts of this program are still in the experimental stage.

$$Fb = 1 + 1.5\,(0.90 - C)$$

Fb = multiplying factor

C = proposed C content

FIG. 2—Formula for calculating boron effect (3).

The general limitations outlined are well established. There are other characteristics of boron less important but worthy of note. Boron changes the critical temperatures of steels only slightly. Usually boron steels are tempered at somewhat lower temperatures than the grades for which they serve as suitable alternates. The proper tempering temperature of 10B45 is 100° to 150° F lower than 9440 to get the identical tempered hardness. (6, 7) To match the same tempered hardness of 4145, 13B45 must be tempered at 150° to 200° F lower. (2)

One of the unique effects of boron on hardenability is that it delays the start of austenite transformation appreciably while delaying the completion of transformation only slightly. (2).

Some of the troubles occasionally experienced in the early use of boron steels have been overcome. The tendency of grain coarsening of ferro-boron treated steels has been licked by using more aluminum in the ladle. At times in the past the last few ingots of a heat showed a lower hardenability. This fading seldom, if ever, occurs today. Experience so far with some of the new boron carburizing grades indicates these steels produce more distortion in a heat treated part than did the old grades. However, it appears that the distortion is more consistent in the new grades and in the long run may be easier to handle. Very recently the manganese contents of a few of the new grades have been raised. This change was made to offset the loss in hardenability caused by using lower carbon to combat distortion of heat treated parts.

The actual reporting of boron content in a heat of steel is not the same as with the other elements because the hardenability effect is obtained over a rather wide range of boron content as long as the minimum boron content is present. Also the minute quantities, 0.0005 percent minimum, plus the difficulties of wet chemistry methods make usual methods for boron impractical. The end quench hardenability tests from each heat are usually all the customers have required. Although accurate chemical and spectrographic tests can be made for boron these tests show total boron content. The actual hardenability effect of boron depends on the amounts of soluble boron and insoluble boron in a steel so that a test which shows total boron content per se is not conclusive. For these reasons, it appears that the end quench test is by far the most suitable and more practical test that has been found to date. A metallographic test for boron in hypoeutectoid steels was developed in 1945. (9) A few users are using this test for information purposes only.

The first known use of boron steel was for experimental cockpit armor of airplanes in 1918. Boron was used to a limited extent during War II on various ordnance parts, armor, AP shot, and vehicle applications. Torsion bars for tanks produced by Cadillac and Buick were made from 92B62. One of the oldest applications of boron steels was for diesel crankshafts. These cranks, still being produced today, are made by Park Drop Forge Co., Cleveland, which employs a 50B46 steel, containing 0.20 to 0.35 Cr.

During the alloy shortage of War II Timken-Detroit Axle developed boron steels for truck axles known as Axalloy. These C-Mn steels performed very well. One of the factors which precluded wider use of the Axalloy grades was that the residual chromium, nickel and molybdenum contents kept increasing. A maximum alloy content was later added to Axalloy specifications to keep the hardenability down. When the residual alloys exceeded the maximum the axles hardened all the way through and failures might have occurred had such axles been put in service. Finally the use of these grades were discontinued because too many difficulties were experienced in keeping the residual alloys low enough.

There was no standardized industry-wide boron steel program during War II so that its use was usually a private affair between steelmaker and user. However, WPB and the War Metallurgy Committees did investigate the merits of boron and some definite programs were carried out. (4, 10, 11)

The first government action directed toward the wartime use of boron-treated steels took place in 1942. Following the request of the War Production Board for an investigation of the merits of "needled steels" in the interest of alloy conservation, two independent surveys were conducted for the purpose of col-

lecting, assembling and interpreting all available data on steels of this type. One survey was directed by a subcommittee of the War Metallurgy Committee and the other by a joint committee consisting of representatives of steel manufacturers, special addition agent manufacturers and steel consumers.

13T45 IS GOOD ALTERNATE FOR WD4145

In January 1943, a new committee on needled steels was formed consisting of representatives of steel manufacturers, steel consumers and the Armed Services, and a testing program was initiated which later became known as Caterpillar Project. (12) Another program known as the Ordnance Project was instigated in July 1943.

The testing program for the Ordnance Project was planned and conducted under the direction of the Subcommittee on Special Addition Agent Steels of the SAE Iron and Steel Committee with the assistance of representatives of AISI, WPB, Army Ordnance, War Metallurgy Committee and others who participated.

As a result of these programs, some fundamental conclusions were reached and had the war lasted much longer, more boron steels would have been used. The conclusions of this work appear below; (4) at that time, the letter "T" was used to signify boron instead of the "B" designation now in use.

*The seven commercial addition agents tested, when added in the more favorable amounts, were all substantially equally effective in improving the properties of the 0.45 C—1.50 Mn steel. This steel, so treated (13T45), was practically equivalent in hardenability (J-50) and mechanical properties to the SAE 4145 with the exception of the lower notched toughness of the former steel at relatively low hardness (*Rc *40 or less), especially when tested at low temperatures.*

The 13T45 steel proved suitable for army truck parts (torque rod end pin), specified to be made of WD 4145, with respect to both the quality of the finished part and the fabricating properties of the material. The 13T45 steel also proved equal, or superior to WD 8745 for M-1 rifle components under performance tests, and compared favorably with WD 8745 in fabricating properties.

Special laboratory tests were conducted on truck and tractor axle shafts and truck steering knuckles, comparing the 13T45 with SAE, WD and NE alloy steels, but the results were such that no definite conclusions could be drawn.

In fabricating properties, including forging, heat treating, cold drawing and machining, the 13T45 steel was substantially equivalent to the SAE 4145 and similar WD and NE steels, with the possible exception of machining.

BORON STEEL LIMITATIONS

1. More boron content is needed in the lower carbon grades to obtain the maximum hardenability effect.
2. Too much boron may cause the steel to become hot short. In high carbon steels this maximum is about 0.005 pct B; in low carbon grades the maximum is 0.008 pct B.
3. Effect of boron on hardenability decreases with increasing carbon up to about 0.90 C. Above this carbon, boron has no further effect on hardenability.
4. Maximum benefits of boron are only manifest on parts that are thoroughly quenched and drawn as the final heat treatment. A fully tempered martensitic structure is required for optimum properties in the heat treated part.

5. The former rich alloy grades are often used with good results in parts larger than the standard critical cooling section recommended for those grades. If high impact properties are needed in the core, it is not advisable to do this with boron steels.
6. More precise control in heat treating is required with boron steels than with those which were made foolproof by high alloy content.
7. More distortion is likely in heat treating carburized boron steels.

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NOTE: The following letter, received by the author and objecting to the emphasis placed on the limitations of boron steels, is reprinted to indicate some disagreement with Mr. Brown's article.

Dear Mr. Brown:

"For some time I have been planning to write you and tell you that you have done a very excellent job in writing up the boron steels in Part I of your Iron Age series. The work is well organized and presented. Articles such as these, if they are read carefully, will do a great deal to illustrate the usefulness of the boron steels. We do not, however, agree with your idea of emphasizing the limitations shown in Part I. Some of them, while true, are not of great enough importance to be highlighted. I have discussed the statement of the boron steel limitations with several other metallurgists and find that none of us agree with you in this respect. However, others probably do.

1. More boron needed for low carbon grades: True. Possibly 6 lb. per ton instead of 4 lb. per ton of the addition agent. This is of trivial importance and should not be mentioned.

2. Too much boron makes steel brittle: True. Increasing carbon or manganese by 2½ times will also make steel brittle. There is no excuse for any steelmaker putting in 2½ times the specified amount of any element. This is not a valid objection to boron.

3. Effect of boron nil at or above 0.90 C: This seems to be true. However, boron does increase hardenability at about 0.75 C or less.

4. Maximum benefits of boron only obtained on fully quenched and drawn parts: True. It is also true that the maximum benefits of any alloy are only obtained when parts are fully martensitic before tempering. However, there are very few commercial parts which show complete martensite before tempering. Some metallurgists believe it is only necessary to have 90 pct martensite at the ¾ radius on most parts that are subject to bending and torsion.

5. Refers to the fact that boron may not give high impact strength in the core. This statement is true but probably causes needless fears. It should be emphasized that we are not talking about withstanding impacts in service, but are talking about notch sensitivity in the center of a heat-treated piece of a heavy section. Aside from AP shot and armor plate, we know of no other examples calling for high notch sensitivity in the center of large sections. Most metallurgists in the automotive and tractor industries do not consider it necessary to employ impact tests on specifications. Those who use them have carefully established reasonable minimums.

6. We do not know any justification for the statement that more precise control of heat treating is required for boron steels. This probably means that the boron steels have a wider spread in the hardenability than other steels. The mill records we have seen to date indicate no wider spread for boron steels than other alloys. It may be true that different heat treatments are needed, but we do not believe it is true that closer control is required.

7. The statement that more distortion is likely in heat treating boron steel parts should be qualified to say that more distortion of carburized and hardened boron steels has been reported, but probably will be overcome by the use of a lower carbon steel.

I wish to say again that you have demonstrated a great ability in writing technical material, and we certainly hope that you will continue to write up the boron steels. You can be of great assistance to us in promulgating the information developed by the boron division. We believe it would be well, however, to present both sides of debatable questions. I hope that you will not be offended at the criticisms we have offered, which we hope are constructive."

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Supervisor of Materials Engineering
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BILGE STEAM SMOTHERING SYSTEM TEST

HAROLD L. COFFIN

THE AUTHOR

served apprenticeship for Engine Draftsman at the Wm. Cramp & Sons, Ship & Engine Building Company, Philadelphia. Attended Drexel Institute of Technology and Temple University. Registered Engineer in the State of Pennsylvania. Test and Trials Engineer at the Philadelphia Naval Shipyard since 1934. Author of several published technical papers on subjects allied to the testing of marine equipment.

INTRODUCTION

Due to lack of basic data, the design of steam smothering systems for Class B (flammable liquids) fires in machinery spaces of a vessel is generally based on individual judgment as to size, spacing and length of steam spray piping. The steam supply is usually taken from the auxiliary steam main. In order to obtain qualitative data for use in the proper design of such systems, a series of tests was conducted at the Philadelphia Naval Shipyard under the general direction of the Bureau of Ships.

The test was divided into two parts. The first part, or *TEST ABLE*, was essentially a flow-rate pressure-drop study to determine the optimum length of spray pipe that would produce an effective steam spray over its entire length with several given pressures at the inlet end. In order to limit the variables which would otherwise be imposed, the size of spray piping, size of orifices and angle of spray were fixed. The second part, or *TEST BAKER*, was made to determine and evaluate the effectiveness of a smothering system (designed from data obtained by *TEST ABLE*) in extinguishing a bilge fire. It was also decided to obtain some comparison between steam, water spray and foam in extinguishing Class B fires under identical conditions. The following is a thumbnail review of the theory of, and media used to extinguish, Class B fires. When flammable liquids are ignited, vapor is released at the surface. The vapor burns, and (except for a shallow layer) the liquid is relatively cool. In other words, the fire is on top. Therefore, if the supply of oxygen can be removed and/or the temperature reduced, the fire will be extinguished. This can be accomplished by providing a "smothering blanket" of steam, foam, carbon dioxide or water fog. Such a blanket reduces the oxygen content immediately above the surface and also materially reduces the vapor temperature.

DESCRIPTION OF TEST ABLE

TEST ABLE determined the limit of length of a single ¾" IPS perforated spray pipe that could be fixed to a supply header to give effective spray with supply steam header pressures of 80, 100, 120 and 140 psi (g). The limit was assumed to have been reached when either of the following conditions existed:

(a) Pressure at remote end of spray pipe was less than sixty (60) percent of the inlet pressure.

(b) The length of plume or effective linear coverage of steam spray from the nozzles at the remote end was less than seventy-five (75) percent of those at the inlet end.

The test equipment consisted of a locomotive supplying saturated steam through a lagged 2½" IPS standard weight, black steel, header, to one ¾" IPS standard weight, black steel, perforated spray pipe of various lengths. Steam flow was measured by an orifice flow meter. Wood battens, graduated in 12" increments, were used to measure the steam plumes. Figure 1 is a diagrammatic arrangement of the test equipment. The length of perforated spray piping varied from 32'-6" to 9'-6". Pipe perforations consisted of ⅛" diameter drilled holes. The holes were drilled in two rows 150 degrees apart, alternately staggered to provide an orifice every lineal 6" and a steam discharge 75 degrees each side the vertical centerline.

Prior to recording data, header and pipe drains were opened and steam allowed to issue for three minutes in order to prevent excessive condensation. Header pressures were obtained by regulating the globe stop valve in the header. Each test was of fifteen minutes' duration and data were recorded ten minutes after start of test. Data recorded are shown in Figure 2.

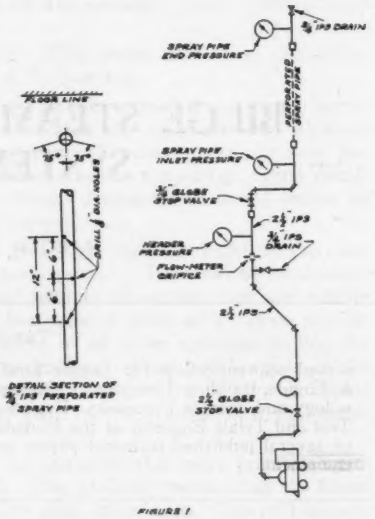

FIGURE 1

The recorded length of steam plumes from the spray pipe listed in Figure 2 is not reliable. The actual length, as determined by several observers, varied from twelve inches for short plumes, to as much as six feet for long plumes.

TEST NUMBER	NUMBER OF ¾" IPS SPRAY PIPE	HEADER PRESSURE (GAUGE) STEAM INLET PRESS. PSI (g)	SPRAY PIPE INLET PRESS. PSI (g)	LENGTH OF PERFORATED PLUME	LENGTH OF SOLID PLUME	POUNDS STEAM PER HOUR	AIR DATA DRY BULB °F	AIR DATA WET BULB °F	AIR DATA % REL. HUMID. ???
1	66	80	67	8	16'-0"	9'-0"	990	70	66	81
2	57	80	44	11	6'-0"	4'-0"	920	68	66	90
3	47	80	48	16	4'-6"	3'-6"	920	69	67	46
4	37	80	50	24	5'-0"	3'-6"	900	72	68	82
5	28	80	54	36	10'-6"	8'-6"	840	68	62	71
6	19	80	69	62	5'-0"	4'-0"	760	74	65	61
7	66	100	60	11	17'-0"	5'-0"	1100	68	66	90
8	57	100	60	16	5'-0"	5'-0"	1080	68	68	86
9	47	100	60	24	5'-6"	4'-0"	1040	71	69	68
10	37	100	68	32	6'-6"	7'-6"	1020	66	60	80
11	28	100	70	49	9'-0"	7'-0"	1000	67	62	76
12	19	100	78	65	5'-0"	4'-0"	780	77	66	46
13	66	120	73	16	6'-6"	4'-0"	1320	68	64	80
14	57	120	78	22	6'-6"	6'-0"	1300	70	58	46
15	47	120	78	31	6'-6"	6'-0"	1280	70	57	42
16	37	120	80	42	7'-6"	7'-6"	1240	65	62	84
17	28	120	85	61	10'-6"	9'-0"	1200	68	64	81
18	19	120	98	81	4'-0"	4'-0"	1060	77	66	56
19	66	140	85	20	6'-6"	5'-0"	1440	68	63	78
20	57	140	85	28	8'-6"	8'-0"	1440	68	63	86
21	47	140	83	36	6'-6"	5'-6"	1420	70	58	46
22	37	140	83	50	6'-6"	6'-0"	1400	63	61	90
23	28	140	102	72	5'-0"	5'-0"	1360	76	66	66
24	19	140	118	95	5'-0"	5'-0"	1280	77	66	56

FIGURE 2

Figures 3 to 9 inclusive, are photographs showing steam flow at various header pressures and number of orifices.

Figures 7 and 8 show the effect of humidity on steam flow. Data for Figure 7 are listed under TEST 20 and for Figure 8 are listed under TEST 21 in Figure 2.

From the results of TEST ABLE it was decided to design the piping installation from the following:

Header Press.	*No. of Orifices*	*Spray Pipe Inlet Press.*	*Spray Pipe End Press.*	*Lb. Steam Per Hour*
80	28	54	36	840
100	28	70	49	1000
120	28	85	61	1200
140	37	83	50	1400

The above data are based solely on pressure drop; i.e., spray pipe end pressure being greater than sixty per-cent of the inlet pressure.

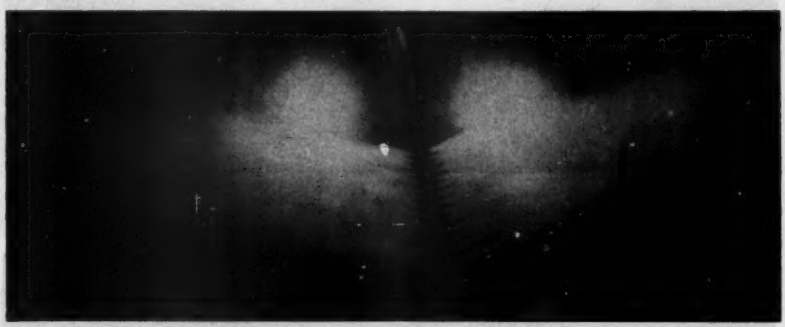

FIGURE 3.

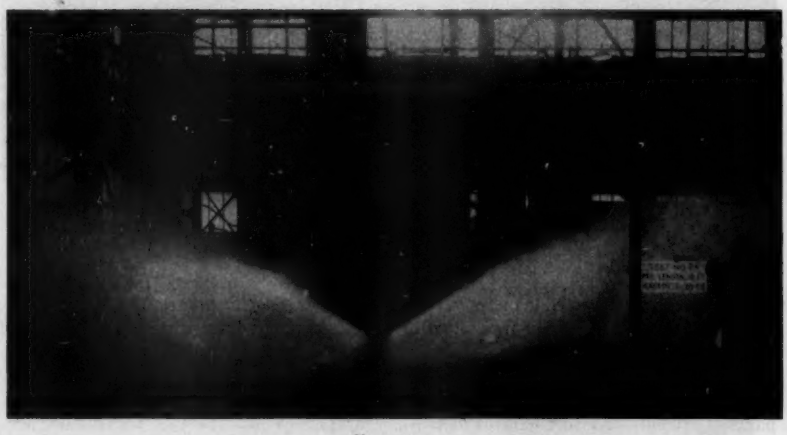

FIGURE 4.

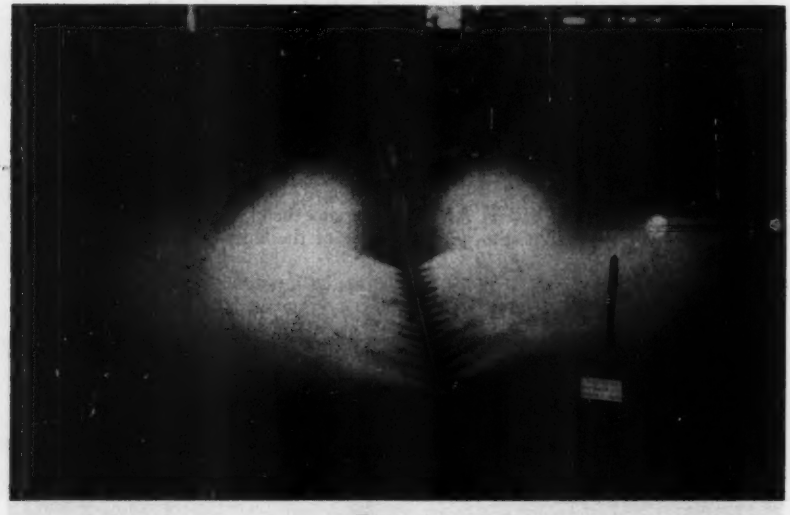

FIGURE 5.

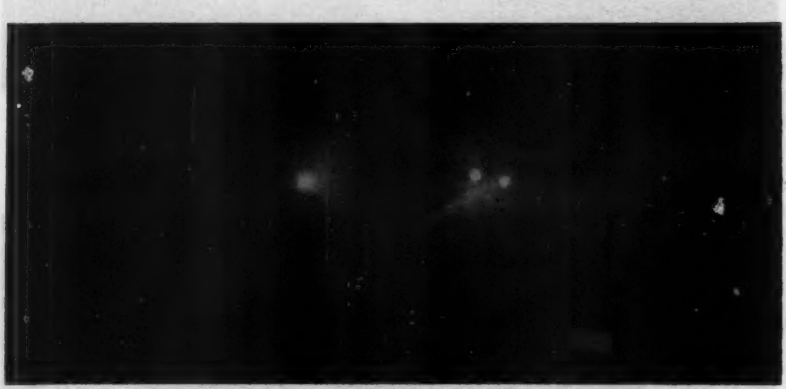

FIGURE 6.

DESCRIPTION OF TEST BAKER

This test was conducted in one compartment of a 500-ton steel coal barge afloat in a dry dock. Saturated steam was obtained from the shipyard steam system through an orifice flow meter, and six (three in parallel) twenty-five feet lengths of two-inch flexible steam hoses to the fixed spray piping installation on the barge. Compartment temperatures were obtained by using an iron-constantan thermocouple and potentiometer pyrometer. The comparatively low compartment temperatures were due to the ventilation provided by the access holes in the deck.

FIGURE 7.

The test compartment was fitted-out to simulate a typical engine room by placing flat steel plates at the approximate normal locations of main engines, reduction gear and various other equipment.

The compartment was filled with water to a depth of ten inches. A three-inch bed of Navy Special fuel oil was laid on the water. This oil bed thickness was maintained throughout the tests. The oil was fired by placing gasoline on the oil and igniting with a torch. The fire covered approximately 1300 square feet. Mechanical foam (liquid type) was fed to the spray piping by a high capacity foam generator through two 2½" fire hoses to the header on the barge for the foam tests.

Figure 10 illustrates the method of outfitting the compartment for the fire tests.

FIGURE 8.

FIGURE 9.

All test fires were given a "pre-burn" period to enable the oil surface to be thoroughly ignited prior to using the extinguishing media.

The following tests were conducted:

Test Number	*Type of Test*
1	Steam-Barge header pressure = 62 psi (g)
2	Steam-Barge header pressure = 60 psi (g)
3	Steam-Barge header pressure = 52 psi (g)
4	Steam-Barge header pressure = 44 psi (g)
5	Steam-Barge header pressure = 39 psi (g)
6	Liquid foam through spray piping
7	Water through L11 waterfog heads
8	Water through L11 waterfog heads
9	Liquid foam through L11 waterfog heads

Figure 11 tabulates the results of the above tests.

Test Number 1

Air data—67° FD.B. 64° FW.B. 85% relative humidity. A pre-burn time of 2 minutes 50 seconds was allowed. The fire completely covered the oil surface and compartment temperature reached 870° F. Header pressure—62 psi (g).

The fire was completely extinguished in one minute twelve seconds, using 240 pounds of steam (equivalent to 29 gallons of water).

Test Number 2

Air data—72° FD.B. 71° FW.B. 98% relative humidity. A pre-burn time of 2 minutes 45 seconds was allowed, and compartment temperature reached 800° F. Header pressure—60 psi (g).

The fire was completely extinguished in 55 seconds, using 190 pounds of steam (equivalent to 23 gallons of water).

Test Number 3

Air data—75° FD.B. 69° FW.B. 73% relative humidity. Pre-burn time —3 minutes 04 seconds. Compartment temperature—825° F. Header pressure —52 psi (g).

Fire completely extinguished in 1 minute 06 seconds with 172 pounds of steam (equivalent to 21 gallons of water).

Test Number 4

Air data—78° FD.B. 71° FW.B. 70% relative humidity. Pre-burn time —2 minutes 05 seconds. Compartment temperature 800° F. Header pressure —44 psi (g).

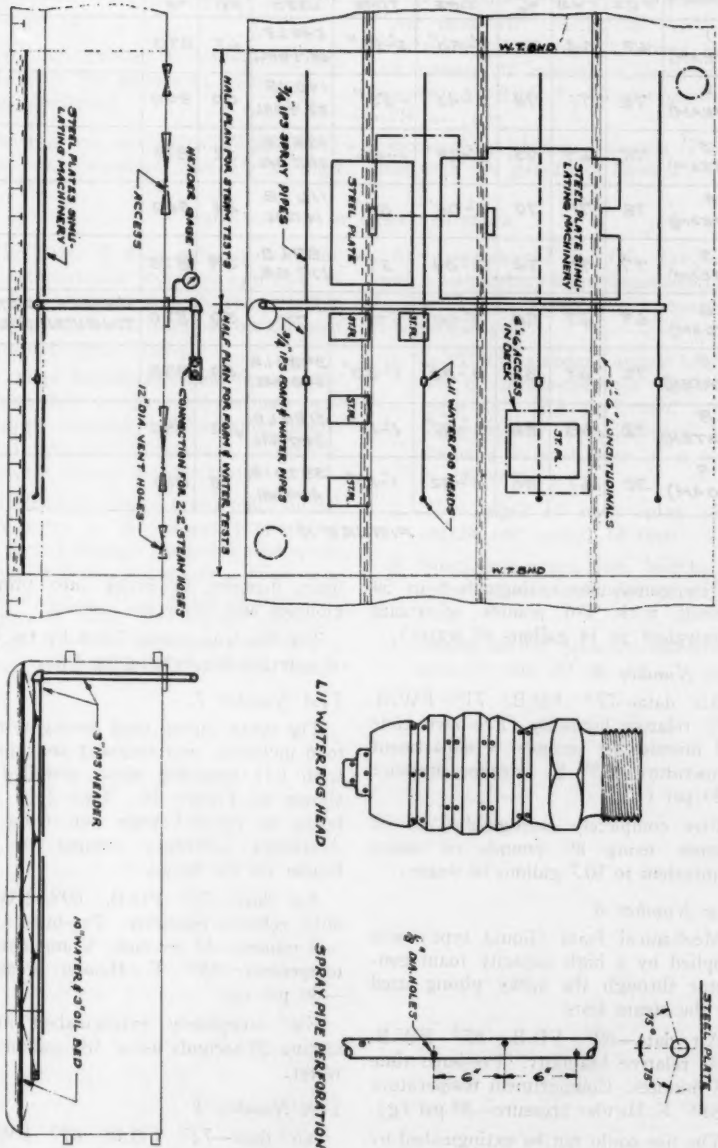
W.T.BHD
STEEL PLATE SIMULATING MACHINERY
2'-6" LONGITUDINALS
4'-6" ACCESS IN DECK
3T. PL.
STEEL PLATE
STR.
LII WATER FOG HEADS
W.T.BHD
¾" I.P.S. SPRAY PIPES
¾" I.P.S. FOAM & WATER PIPES
HALF PLAN FOR STEAM TESTS
HALF PLAN FOR FOAM & WATER TESTS
STEEL PLATES SIMULATING MACHINERY
ACCESS
MERCER GAUGE
CONNECTION FOR 2-2" STEAM HOSES
12" DIA. VENT HOLES
3" I.P.S. HEADER
10" WATER & 3" OIL BED
LII WATER FOG HEAD
SPRAY PIPE PERFORATIONS
⅜" DIA. HOLES
6"
6"
STEEL PLATE
75°
75°

FIGURE 10

TEST NUMBER	AIR DATA °F D.B.	AIR DATA °F W.B.	REL. HUMID. %	PRE-BURN TIME	EXTIN. TIME	ACTUAL QUAN. USED	HEADER PRESS. PSI	COMPT. TEMP. °F	REMARKS
1 (STEAM)	67	64	85	2'-50"	1'-12"	240 LB. 28.9 GAL.	62	870	
2 (STEAM)	72	71	98	2'-45"	55"	190 LB. 22.9 GAL	60	800	
3 (STEAM)	75	69	73	2'-29"	1'-06"	172 LB. 20.7 GAL.	52	810	
4 (STEAM)	78	71	70	2'-05"	54"	116 LB. 14 GAL.	44	800	
5 (STEAM)	77	71	74	3'-04"	52"	89 LB. 10.7 GAL.	39	825	
6 (FOAM)	69	67	90	3'-00"	—	—	30	830	UNABLE TO EXTINGUISH FIRE
7 (WATER)	72	69	86	3'-30"	1'-30"	3980 LB. 480 GAL.	80	835	
8 (WATER)	72	69	86	3'-15"	1'-15"	3180 LB. 380 GAL	72	790	
9 (FOAM)	70	67	86	3'-02"	1'-17"	3320 LB. 400 GAL.	78	810	

FIGURE 11

Fire completely extinguished in 54 seconds with 116 pounds of steam (equivalent to 14 gallons of water).

Test Number 5

Air data—77° FD.B. 71° FW.B. 74% relative humidity. Pre-burn time —3 minutes 04 seconds. Compartment temperature 825° F. Header pressure —39 psi (g).

Fire completely extinguished in 52 seconds using 89 pounds of steam (equivalent to 10.7 gallons of water).

Test Number 6

Mechanical foam (liquid type) was supplied by a high capacity foam generator through the spray piping used for the steam tests.

Air data—69° FD.B. 67° FW.B. 90% relative humidity. Pre-burn time —3 minutes. Compartment temperature —830° F. Header pressure—30 psi (g).

The fire could not be extinguished by this method due to the inability of the foam mixture to break into minute globules and cover the surface.

The fire was extinguished by the use of external fire-fighting facilities.

Test Number 7

The spray piping used during tests 1 to 6 inclusive, was removed and piping with L11 waterfog heads installed as shown in Figure 10. Two 2½" fire hoses in parallel were run from an American LaFrance pumper to the header on the barge.

Air data—72° FD.B. 69° FW.B. 86% relative humidity. Pre-burn time —3 minutes 30 seconds. Compartment temperature 835° F. Header pressure —80 psi (g).

Fire completely extinguished in 1 minute 30 seconds using 480 gallons of water.

Test Number 8

Air data—72° FD.B. 69° FW.B. 86% relative humidity. Pre-burn time

—3 minutes 15 seconds. Compartment temperature—790° F. Header pressure —72 psi (g).

Fire extinguished in 1 minute 15 seconds with 380 gallons of water.

Test Number 9

For this test, mechanical (liquid type) foam was supplied through a high capacity foam generator, two 2½" fire hoses and fixed L11 waterfog heads.

Air data—70° FD.B. 67° FW.B. 86% relative humidity. Pre-burn time —3 minutes 02 seconds. Compartment temperature—810° F. Header pressure —78 psi (g).

Fire extinguished in 1 minute 17 seconds with 400 gallons of liquid.

CONCLUSIONS OF COMBINED TESTS

The Class B fires generated in the test compartment were extinguished in the least time and with the smallest quantity of liquid, using steam through perforated spray piping.

A bilge flammable liquid fire can be extinguished using water through permanently installed L11 waterfog heads. The liquid weight, however, is greatly in excess of that using steam.

Mechanical (liquid type) foam mixed with water in the proper proportions and forced through the perforated spray piping, will not extinguish a bilge fire. This is due to the inability of the foam mixture to break into minute globules. It will, however, extinguish an oil fire if used with fixed L11 waterfog heads.

As demonstrated in the foregoing tests, a satisfactory bilge steam smothering system may be designed to conform to the following:

1. A 3" IPS, standard weight header will be satisfactory for supplying eight ¾" IPS perforated spray pipes having 35 orifices of ⅛" diameter in each pipe, or a total of 280 orifices.
2. The length of each spray pipe should not exceed 18 feet.
3. Spacing between two lengths of spray pipes should not exceed 8 feet.
4. Steam header pressure should be greater than 40 psi (g).

A SURVEY OF PRESENT METHODS OF SUPERHEATER CONTROL

ACKNOWLEDGMENT

This article appeared in the August and September 1951 issues of "The Marine Engineer and Naval Architect." It consists of extracts from a report recently made available for publication by the British Shipbuilding Research Association. The author is LT. CDR. (E) J. E. J. NOTTIDGE, RN, (Ret'd).

The trend of modern steam-installation design is ever in the direction of increased efficiency, and this naturally leads to the use of higher steam temperatures and pressures. As a direct result, materials, both in the boiler and turbine, are being stressed nearer to their mechanical limits, the more so as the mechanical properties of metals deteriorate rapidly as the working temperature is increased. It is thus found necessary to exercise some definite control over the degree of superheat which the boiler is producing. Such a system of control should perform the following functions:

1. It must prevent the steam temperature from exceeding the allowable maximum at all times, thus obviating the possibility of failure of both boiler and turbine components due to overheating.

2. It should enable the degree of superheat to be kept to the allowable maximum over a wide range of evaporation, thus maintaining the efficiency of the turbines and preventing excessive wetness of steam at low powers.

3. It should provide sufficient range and speed of control to cope with variation of superheat temperature brought about by changes in the following:

(*a*) Temperature of feed water to the boiler.
(*b*) Percentage of excess air required for complete combustion.
(*c*) Percentage of moisture in the steam entering the superheater, due to "carry over."
(*d*) Percentage of saturated steam being used to feed auxiliaries, etc.
(*e*) Temperatures of piping and machinery during warming up.

4. When the astern turbine is not designed to stand the ahead superheat temperature, it must enable the degree of superheat to be lowered (several hundred degrees possibly) at a specified percentage of full power; the more so as, when going astern, feed heat is usually cut off and the boilers are given liberal excess air, both factors tending to increase the superheat temperature.

5. It should enable the steam conditions to be kept suitable for harbor machinery, when main engines are shut down and the boiler is being steamed at limited output.

SUPERHEATER CONTROL

SUPERHEATER CHARACTERISTICS

Heat transfer to superheaters may be effected either by convection or radiation, and superheaters are designed to receive heat by either or both of these methods. The radiant superheater produces a greater degree of superheat at low powers, and the temperature falls off as the steam flow through the elements increases; thus the superheater has what is known as a falling characteristic. The convection superheater, on the other hand, has a rising characteristic, and the degree of superheat increases, up to a point, as the boiler output increases. By combining a radiant and a convection superheater, or, as is very usual, by placing a convection superheater near enough to the furnace so that the passing gases are hot enough to radiate an appreciable quantity of heat to the elements, a characteristic that is nearly flat may be produced. It is usual with a superheater so positioned for the convection side to predominate, so that a slightly rising characteristic is obtained. Such designs, without provision for manual control by the operator, do not fully meet the requirements already referred to; but they have proved adequate in service for machinery installations of low and moderate power where the astern turbines are capable of taking steam at 650 deg. F., and indeed are the usual choice for cargo vessels of moderate power. They have the great merit of simplicity of construction and operation.

With higher steam temperatures, however, where the margin of safety is smaller and where the requirements already enumerated must be more stringently met, it is generally accepted that some additional form of control must be provided. The various means for achieving this may conveniently be grouped under five headings:

1. Damper control of the gas flow over the superheater elements.
2. Cooling the superheated steam by contact with saturated steam or water after leaving the superheater.
3. Separately fired superheater.
4. Two-furnace boiler design.
5. Miscellaneous.

The purpose of this paper is to survey the existing methods of control and to compare, as far as is possible, their advantages and disadvantages. It is worthy of note, however, that there is a school of thought, especially in America, who favor a simple boiler with a flat superheat temperature characteristic. Since astern turbines are now being designed to take full ahead superheat temperature, they hold the opinion that simplicity, lightness, and saving in space of such a single-furnace boiler will offset the arguments generally presented for the control of superheat.

DAMPER CONTROL OF GAS FLOW

One of the simpler forms of superheat control is that of varying the gas flow over the elements by the manipulation of suitable dampers. This system can effectively be applied to certain boilers whose construction renders them suitable. One of the simpler applications of this system is the two- or three-drum boiler with a divided gas flow around the steam drum and a superheater fitted in one tube bank only. A "minimum opening" damper is fitted in each uptake so that the proportion of gases flowing up each side may be

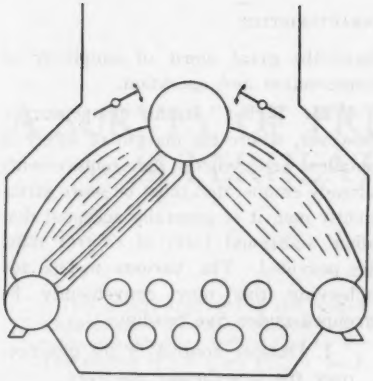

Fig. 1. Double damper control

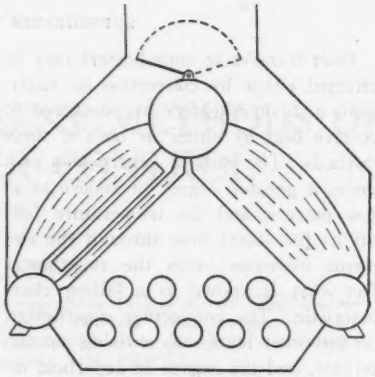

Fig 2. Single damper control

varied; as one damper is opened, the other is closed (Fig. 1). Alternatively, over the steam drum a large unbalanced damper may be fitted which may be moved to obstruct the flow of gas on one side or the other (Fig. 2). This system is modified in some designs having two superheaters of different heating surfaces, one on each side of the boiler, the dampers being independently operated; this restricts the range of control. Such a system would be designed to give full superheat at full load with both dampers open; but to increase the superheat temperature at lower powers, a greater proportion of gas would be deflected over the superheater with the larger heating surface.

In the horizontal cross-drum type of boiler, which often has a number of gas passes, the superheater elements are situated in the divided first pass. A suitable unbalanced damper enables the flow of gases over the superheater to be restricted (Fig. 3). It is to be expected that the restriction of gas flow caused by the use of dampers would show an appreciable drop in efficiency. Stillman, however, quotes the figures taken from the trials of a U. S. destroyer, fitted with a three-drum boiler and damper control, which show the loss to be small. These figures show good control of degree of superheat temperature from 30 deg.—200 deg. F at 80 percent full power, and the maximum variation in efficiency at this evaporation and over this range of temperature was less than 2 percent.

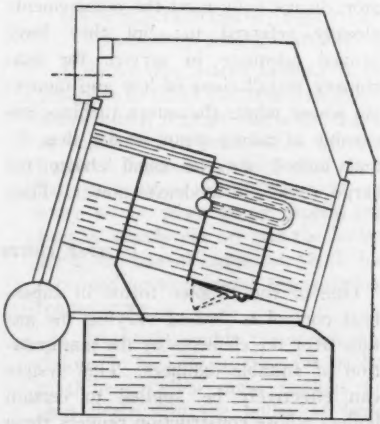

Fig. 3. Damper control as applied to a multi-pass header-type boiler

DAMPER MATERIAL

This type of control in its most straightforward form is satisfactory for temperatures up to 800 deg. F. where a very wide range of control is unnecessary. For gas temperatures in way of the dampers in excess of 800 deg. F., special attention must be paid to the material used and the damper design because of the weakness of materials at these temperatures. It is stated by some that trouble has arisen with dampers at higher temperatures due to warping. It is thought, however, that if due consideration be given in the design stage to the material used, and to the dimension and clearances, this should not occur. Some designers have overcome warping by fitting several small dampers in the form of a venetian shutter in place of a single large damper. This provides a better path for gas flow, but entails additional cost and many extra moving parts, which it is expedient to avoid if possible when working at high temperatures and where efficient lubrication in service is liable to prove difficult.

ADVANTAGES OF DAMPER CONTROL

The damper control system is limited both in range of control and in application. Where these considerations can be overcome it has the overwhelming advantages of being simple in design, economical in weight and space, cheap in outlay and maintenance, and maloperation will not endanger the elements. It is consequently popular in steam installations where elaboration is to be avoided.

STEAM COOLING BY SATURATED STEAM OR WATER

A widely used method of control is that of cooling the steam after it has received superheat. This is frequently an incomplete method of control, for although the temperature of the steam may be lowered at will, it cannot be raised at low powers unless a radiant superheater be used. The mechanism by which this cooling is achieved is generally referred to as a de-calorizer or attemperator, and may be brought about:

(*a*) By spraying water into the steam.
(*b*) By passing the steam over a water-cooled surface.
(*c*) By adding saturated steam to the already superheated steam.

In the first case water is sprayed into the steam to cool or humidify it, and is generally introduced as a very fine spray. Some further mechanical means of mixing the water and steam is found necessary to produce a homogeneous mixture free from stratification. This mechanical mixing is achieved either by whirling the mixture, the steam being introduced tangentially into the mixing chamber, or by impingement of the mixture on suitably designed baffles. The latter method produces a greater resistance to flow but better mixing. Baffling requires more space and hence greater weight than the whirling method.

SPRAYING NOZZLES

The design of the spraying nozzles presents some difficulty, owing to the problem of producing nozzles giving efficient atomization over a wide range of output. This is generally overcome by using a series of nozzles, the number in use being varied according to the amount of water to be sprayed. The use of nozzles with too small a bore renders them susceptible to choking. The injection system is fitted after the superheater, but should be installed fac-

ing upstream at least 60 pipe diameters before the turbine, to obviate the risk of water entering the latter. If the superheater is in two parts, as is sometimes the case, the water is injected after the first superheater. Difficulty is encountered here in effecting an even distribution of the moisture content over the whole bank of superheater tubes. Heated feed water is most commonly used for injection as this ensures that the oxygen content is kept to a minimum. It is usual with this system to fit a thermostatic control to regulate the water spray. The advantages of this system are that it is automatic in operation, is of moderate initial cost, and makes only moderate demands on weight and space. Its efficient functioning, however, is entirely dependent upon reliable operation of the automatic system for controlling the injection rate, and failure would probably lead to damage of the superheater elements or machinery. Moreover, a supply of pure feed water is essential if all chance of deposition in superheater elements and turbines is to be avoided. It is presumably because of these requirements that this system is not at present popular in marine practice, except as a means of providing desuperheated steam for auxiliaries.

SURFACE ATTEMPERATORS

There are several methods of cooling the superheated steam by passing it over water-cooled surfaces. The principle of nearly all these is the cooling of a certain percentage of the superheated steam by passing it through an attemperator. Control is exercised by varying this percentage from zero up to a maximum. The simplest form of this type of attemperator consists of a pipe making several passes through the steam drum beneath the working water level. After leaving the superheater, a certain percentage of the superheated steam may be passed through this pipe where it relinquishes heat to the water, causing increased evaporation. The area of heating surface which can be got into the space available is not large, and so the degree of control possible is not great. This system is popular in plants where low superheat temperature is required, being cheap and foolproof in operation and requiring little maintenance. For larger plants with higher superheat temperatures, the space available in an already overcrowded steam drum is inadequate.

U-TUBE PATTERNS

Another type of surface attemperator takes the form of U-tubes in a shell, the latter having steam and water connections to the boiler. A proportion of the superheated steam passes through the U-tubes, evaporating the water. This type is similar in principle to the internal pipe, but being situated outside the boiler it may be so constructed that the heating surface is as large as is necessary to give the degree and range of control required. It is more costly, more bulky and heavier than the foregoing type of attemperator. It is popular, however, in marine work because of its simplicity and ease of operation and upkeep. This type of attemperator has also been designed to use feed water as the cooling medium. This enables a smaller design to be used on account of the greater difference in temperature between steam and coolant, and also simplifies boiler mountings. Its main disadvantages would appear to be the resultant fluctuations of feed-water temperatures and the fact that maximum feed temperature cannot be achieved with high superheat.

There is a further type of attemperator, which although similar in construction to the U-tube design, effects temperature control on a different principle (Fig. 4). All the superheated steam passes through this attemperator, and temperature control is achieved by varying the surface exposed for heat transfer. This design consists of a vertical cylindrical shell placed level with the boiler and alongside it. The shell is fitted with internal vertical tubes, over which the superheated steam flows, suitably baffled; the tubes are connected to the boiler by steam and water connections. In the water connection to the boiler is fitted a fine thread sensitive valve, which is used to control the outgoing superheat temperature.

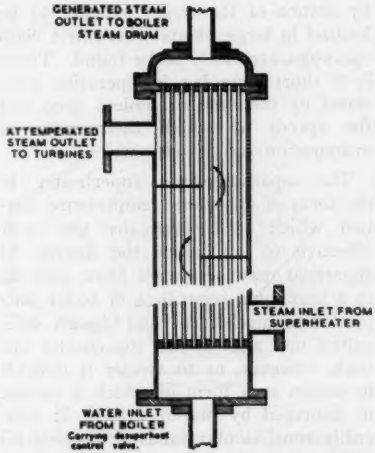

Fig. 4. Vertical evaporative type of attemperator

METHOD OF CONTROL

Control of the superheat temperature is brought about as follows: When the valve is opened, water from the boiler enters the attemperator tubes, and evaporates on coming into contact with the tubes, over which the superheated steam is passing. The water will rise to a height in the tubes such that the rate of evaporation equals the flow of water through the valve in the water connection. By opening the valve the water level is raised and by closing it the level is lowered. The higher the water level in the tubes the greater the cooling effect of the superheated steam and vice versa. This is one of the simplest of external attemperators to control and it therefore enjoys considerable popularity. Control on this principle obviates the necessity for bypass valves and extra piping in the steam raising the superheat temperature due to the period taken to evaporate away the water standing in the tubes; there is no time lag when reducing temperature. This is probably the most popular method of control for low superheat temperatures and moderate powers, notwithstanding the fact that high steam temperatures cannot be obtained at low rates of evaporation without the use of a radiant superheater, which is often inconvenient in marine practice. The initial cost is not excessive nor is the maintenance cost high. It is nearly foolproof in operation and it is not possible to damage tubes by overheating, due to negligence. To obtain wide control for high outputs, however, the attemperator becomes impossibly large.

Where two or more boilers are fitted to a plant, one boiler may be used to generate saturated steam only. This saturated steam mixes with superheated steam from the other boilers. Control is then exercised over the steam temperature by varying the relative rates of evaporation of saturated and superheated generators. This system has only limited application, and is suited to large multi-boiler plants where it is possible to have a saturated steam boiler, which can usually only be justified when reciprocating steam-driven auxiliaries are fitted. This type of control requires well trained operators, but

by nature of its design is only to be located in large plants where first class operators are likely to be found. There is a short time lag in operation governed by the type of boilers used and the speeds at which their rates of evaporation can be altered.

The separately-fired superheater is the form of superheat temperature control which is theoretically the most effective of all, since the degree of superheat may be varied from zero up to a maximum regardless of boiler output. The mechanical and design difficulties met with in this superheater are such, however, as to render it difficult to design in a form in which it cannot be damaged by mal-operation. It generally consists of a furnace, wherein oil fuel is burned, and from which the gases pass at right angles, to flow over the elements of the superheater in order to avoid any direct flame impingement on the superheater elements. It is possible, with the use of suitable materials and adequate steam flow, to design a separately-fired superheater which will withstand flame impingement on the superheater elements. The high cost, however, together with the dangers accompanying the failure of steam flow render this design of little practical value. Consequently it is generally accepted that some means of protecting the elements must be used. This may be done by constructing a furnace sufficiently large for combustion to be complete, and by screening the elements from the flame by suitable refractory baffles. The consequent size and weight of such a construction are so great as to preclude it from general marine use where considerations of space and weight are of primary importance.

AUTOMATIC CONTROL

All types of separately-fired superheater, when oil-fired, may easily be adapted to automatic superheat control, whereby the supply of oil to the furnace is regulated by the outgoing steam temperature, the superheater being by-passed when maneuvering.

The separately fired superheater is frequently incorporated in the boiler and built integral with it (Fig. 5). This form is the more convenient and besides saving weight and space, it also allows for a better arrangement of water tubes to protect the superheater elements. When steaming at low superheat temperatures, however, a considerable area of potential generating, feed heating and air heating surface is wasted. Considerable numbers of this type of boiler have been fitted in high-powered vessels in the U. S. A., but the economic use of such a superheater requires a larger evaporation than is found in all but the largest merchant ships, and it is likely that an additional operator would also be required. These disadvantages together with the added complication of extra steam piping and by-pass valves are generally held to outweigh the advantages of complete and positive temperature control.

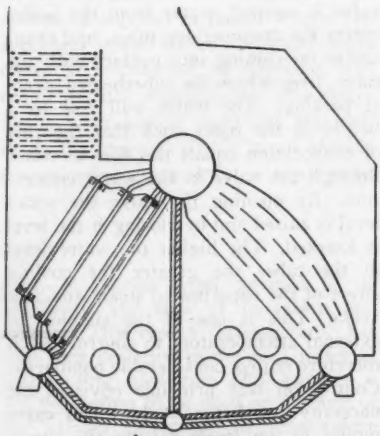

Fig. 5. (left) Integral separately-fired superheater

TWO-FURNACE BOILER

This type of boiler has for some time past been used in the U. S. A. but only lately have its potentialties been realized in this country. A sketch of a typical arrangement is given in Fig. 6, where the steam drum A is connected to the main water drum B by the large tube bank D, and to the intermediate water drum C by the smaller bank E. The superheater elements H are located in the main tube bank D. The tubes M rolled into the header J form a water-wall to the outer furnace G. The gases from the furnace F pass through the main tube bank D and over the superheater elements on their way to the uptake K. Those from the furnace G pass through the intermediate tube bank and into the furnace F where they join the gases from the latter. The gases from the furnace G will be much cooler, when they reach the superheater elements, than will the gases from F, since the former have lost heat by radiation to the water-walls M and by convection to the intermediate tube bank E. By varying the proportion of oil being burned in the two furnaces the temperature of the gases passing over the superheater element and hence the superheat temperature may be controlled. At full power the boiler would produce the designed maximum superheat temperature with all sprayers burning in furnace F and probably all but one in furnace G.

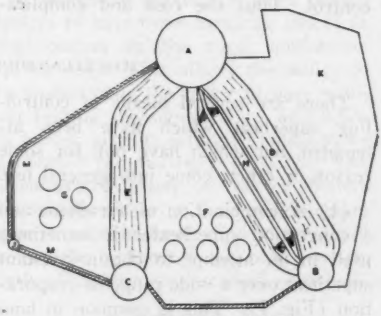

Fig. 6. Two-furnace controlled superheat boiler

This type of boiler gives a very good control of superheat temperature which may be maintained at the maximum down to very low rates of evaporation. It may also be designed to give maximum superheat temperatures as high as the limits of materials will allow. The control of the temperature is quick and positive and it is not easy to damage components by mishandling the boiler; but care must be taken to ensure that if sprayers are alight in furnace F one sprayer *must* be alight in furnace G, otherwise the latter would constitute a stagnant pocket in which explosive gases could accumulate. A small "simmering" sprayer is often fitted in furnace G to overcome this. Since the boiler has two furnaces its operation is not as straightforward as a single-furnace type; thus personnel require some special instruction in its operation. The initial cost of such a boiler is higher than the more orthodox type, although by nature of its larger heating surface it would probably have a higher evaporation than a more usual type of boiler occupying the same space. For the same evaporation, it weighs little more than any other type of boiler in general use having some sort of superheat control.

The cost of maintenance of the two-furnace boiler is higher than that of the more straightforward types, on account of the greater area of brickwork and also the large number of small hand hole doors necessary to give access to the water-wall tubes. The initial cost of the boiler would be high, but the combined cost of a simpler boiler with an external means of superheat control would be little less. This type of control is better suited to higher powered plants, with superheat temperatures of 700 deg. F. and above, because several

burners in each furnace are necessary to obtain a fine degree of temperature control. Thus the cost and complications of such a system do not justify themselves for low power and low temperature plants.

MISCELLANEOUS OTHER METHODS

There are several means of controlling superheat which have been attempted, but which have not, for some reason or other, come into general use.

(1) A combination of a radiant and a convection superheater is sometimes used in an attempt to obtain constant superheat over a wide range of evaporation (Fig. 7). This is common in land practice but is not popular afloat. To incorporate such a system usually means a combustion space, and hence a boiler, larger and heavier for its output than can be tolerated in marine practice. Such a system does not allow for reduction of superheat temperature for maneuvering; and in the event of a sudden "stop" from high power, the flow through the radiant superheater would probably be insufficient to save it from damage.

(2) Radiant superheaters have been fitted in boilers and so placed in relation to sprayers that the radiation to the superheaters will be affected by the individual sprayers in use and the variations in air pressure. This system of control suffers from the disadvantages which accompany the use of radiant superheaters, and also the range of control is limited and further complicated by the number of sprayers which must be in use for a given evaporation. Moreover, it does not allow for reduction of superheat temperature at low powers.

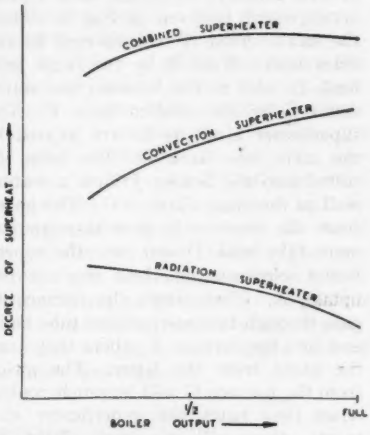

Fig. 7. Typical characteristic curves of convection and radiant superheaters

(3) In the Loeffler boiler, steam circulation is used to absorb heat from the fuel burned. A portion of the superheated steam is taken to the main engines and the remainder to the water-drum for the production of saturated steam. By varying the speed of the steam-circulating pump the proportion of heat for generating and superheating may be varied and so the superheat temperature is controlled.

(4) In a recent marine La Mont boiler the superheat temperature is effectively reduced by introducing cool air into the gas stream at the superheater.

SUPERHEAT CONTROL IN SCOTCH BOILERS

The requirements from Scotch boilers in service may differ somewhat from those of water tube boilers. Although Scotch boilers are frequently installed with sets of turbines, they are as a general rule used in installations of lower power where reciprocating engines are still favored. The search for efficiency, however, has led to the use of reheat in such engines, and it is now common practice to use Scotch boilers delivering steam at a temperature up to 750 deg. F.

This temperature is reduced before reaching the engines, by passing it through the interstage reheater, but some form of control is necessary to prevent it exceeding the maximum. In order to protect the superheaters, it is generally the rule to have all steam generated passing through them. The steam is then controlled to the required temperature, for the main engines, and also to a greatly reduced temperature for the auxiliaries by passing a proportion through some form of attemperator. The auxiliaries are fed with attemperated steam, and the remainder rejoins the superheated steam and cools it to the required temperature.

It is general practice in large Scotch boilers to have three furnaces, and to fit superheaters in the wing combustion spaces only. This allows the boiler to be flashed upon the center furnace without heating the superheaters, and when in harbor permits saturated steam to be generated for auxiliary purposes. The attemperators used on Scotch boilers are usually either of the spray-injection type or take the form of an internal coil below the surface of the boiler water, both of which are described elsewhere.

SUMMARY AND CONCLUSIONS

The practical means of controlling superheat temperature may be divided into four main methods:

(*a*) Damper control of gas flow.
(*b*) Cooling the superheated steam with saturated steam or water.
(*c*) Separately-fired superheater.
(*d*) Two-furnace boiler.

Damper control is the simplest method, and also the cheapest in outlay and maintenance. Its application is, however, limited to certain boilers, and the degree of control is limited. Where the damper is situated in high gas temperatures, complications arise in construction and it is doubtful whether the use of the damper control method is here justified. Dampers are foolproof in operation and best suited to boilers where a wide degree of control is unnecessary. They cause a slight loss in efficiency due to impedance of gas flow, but in well-designed damper systems this is small.

Cooling the superheated steam by water in an attemperator is probably the most universally-used method in marine practice today. Cooling may be effected by the use of either feed or boiler water. A variety of mechanical means are used to achieve the heat transfer. In general the methods are all simple, and straightforward in operation. Unless a radiant superheater (which will give a high degree of superheat at low rates of evaporation) be incorporated in the system, this method suffers from the disadvantage that maximum superheat temperature cannot be obtained at low output. Where cruising at reduced powers is not an important consideration, this is not, however, a great disadvantage. This system costs considerably more to install than dampers, but less than many other methods, and the weight and space occupied are not excessive. It is simple in operation and maintenance.

The separately-fired superheater is the only absolute method of superheat control, and with it any superheat temperature may be obtained at any output. Design considerations make it extremely difficult to construct such a unit which is either small or light, and it is necessarily costly. The separately-fired superheater often requires an extra operator, who must be reasonably skilled since mishandling the superheater can endanger the elements. This superheater only really justifies itself in large installations. The above disadvantages render it unpopular, so that it is infrequently met with in marine work. The integral separately-fired superheater is not sub-

ject to these disadvantages however, but its complication and cost are such as to limit its use to high-powered installations.

The two-furnace boiler strikes the mean between the separately-fired superheater and other methods of control. Its range of control is not as comprehensive as that of the separately-fired unit, but is adequate for marine needs. It is more costly than a straightforward boiler and requires a higher degree of skill in operation than do the simpler systems, but the superheater elements are secure from damage due to mishandling. It can be built within the limits imposed by considerations of space and weight, and its popularity in this country is increasing for medium- and high-power marine installations.

No specific method of control is so lacking in disadvantages as to overshadow the others, but the general trend seems to indicate that, for lower powers and temperatures, the external attemperator is favored because of its simplicity in construction and operation, while at increased powers and temperatures the two-furnace boiler is coming into its own. All methods of control are subject to the human element and, in the event of a sudden "full astern" when the engine and boiler room personnel are not expecting it, the astern turbine is liable to become overheated, unless the boiler watch-keepers are exercising extreme vigilance and are very smart in controlling the superheat temperature.

In conclusion, the question arises whether the cost and complication of obtaining the control of superheat temperature desired is worth the advantages gained thereby. Should the use of astern turbines, capable of withstanding full ahead superheat temperatures become general, as seems likely, then the case for superheat control will be seriously weakened. It should be borne in mind, however, that, although astern turbines can be designed to take full ahead superheat temperature, the superheat temperature often tends to rise when going astern, because of the cutting-off of feed heat and the more liberal supply of excess air. Care must therefore be taken to ensure that astern turbines capable of taking full ahead superheat temperature are not in practice called upon to deal with temperatures in excess of those for which they were designed.

THE SIZE AND SPEED OF TANKERS
Recent Improvements in Construction

ACKNOWLEDGMENT

JOHN LAMB, Manager of the Marine Research and Development Department of the Anglo-Saxon Petroleum Co., Ltd., and R. L. BROWN, Naval Architect of the same department, are the authors of this article, which is an extract from a paper read at the World Petroleum Congress. It was published in the October 1951 issue of "The British Motor Ship."

Never before have there been so many bulk oil-carrying ships under construction and, in view of the constantly increasing application of petroleum and its many and varied products, it is safe to say that this class of ship will form a larger proportion of the world's future shipbuilding than has been the case hitherto, particularly as the average useful life of an oil tanker is considerably less than that of a dry cargo carrier.

To illustrate the phenomenal growth of the world's oil tanker fleet during the past 20 years the number of ships built or on order in January of the years 1930 and 1950 is given below. Tankers of 1,000 tons deadweight and under have not been included but it can be assumed that in the period reviewed there has been a great tonnage increase in this class of oil tanker also.

The figure of 21.0 given as the tanker percentage of the world's gross tonnage in the year 1950 includes the huge American fleet of merchant ships not in active service. If this tonnage had been excluded, tankers at the beginning of 1950 would represent in gross tons 25.5 percent of the world's merchant shipping.

Apart from the introduction of welding, improved living quarters for officers and crew and the provision of aids to navigation and labor-saving appliances, tankers building today are not so very different from what they were 20 years ago, excepting that they are larger and faster. Deadweights of the largest tankers have gone up from 12,000 to 30,000 tons and speeds have increased from 12 knots to 16 knots. In some instances speeds in excess of 16 knots have been adopted, but such ships were designed for special purposes. The Shell Group, for example, own two 18,000-ton d.w. tankers capable of a sustained speed of 18 knots when fully loaded.

	1930	1950
Number of tankers	1,272	2,149
Deadweight tonnage	10,100,000	24,450,000
% of world's shipping, in numbers	8.52	15.1
% of world's shipping, in gross tons	11.69	21.0
Tankers building or on order, in numbers	85	174
Tankers building or on order, in d.w. tons	872,000	2,850,000

PORT FACILITIES

Passenger and dry cargo ships use ports selected because of deep water and other natural facilities, and there is no objection to the docks used by such ships being situated near densely populated areas. In the case of oil tankers, however, the loading terminals are mostly selected because of their close proximity to oil fields, while the discharging terminals must, because of the hazards involved, be situated well away from the ports normally used by shipping and, incidentally, the best harbor facilities. The notable absence of accidents when handling bulk petroleum cargoes, however, is having the effect of gradually eliminating this discrimination.

It was because of these restrictions that tankers built prior to 1939 were comparatively small when compared with those now building. Economics at that time dictated that they should be suitable for world-wide trading and there is no doubt that tankers of about 12,000 tons deadweight drawing from 27 ft. to 28 ft. of water were the most suitable.

Since that time harbor facilities have remained practically unchanged, but the cost to build, repair and man tankers has increased so much that owners of such ships have been compelled to explore every channel likely to result in a reduction of the cost to transport a certain quantity of cargo over a given distance. As the cost to build and operate ships per deadweight ton is progressively reduced as the size of the ship increases, the general desire for larger tankers is understandable.

The principal dimensions of a seaworthy and efficient ship, such as length, breadth and depth, must bear a certain relationship, so that if the length and breadth, which dimensions mainly affect the carrying capacity, are increased, the depth and consequently the draft must be increased in proportion. Moreover, for a given standard of strength the length, beam and depth have very definite ratios if the best is to be made of the steel used in construction.

Although it is mainly the draft which limits the trading flexibility of a ship, the length and breadth also offer restrictions in some ports. For instance, the greatest overall permissible length for tankers navigating the river Seine is 557 ft. Limiting the trading flexibility of tankers by increasing their size means, therefore, that the largest tankers now building will be able to use only a small number of recognized oil ports.

To illustrate how an increase in deadweight results in a reduction of the cost to transport, the costs to carry each ton of cargo on a voyage Curaçao/U.K./Curaçao in tankers of three different sizes having a speed of 11½ knots have been calculated. The deadweights selected are 10,000, 20,000 and 30,000 tons, and giving the calculated operating cost of the 10,000-ton ship the basic figure of 100 it is found that the corresponding figure for the 20,000-ton ship is 66 and 52 for the 30,000-ton ship. Thus, the cost to transport each ton of cargo in the 30,000-ton ship is a little more than half the cost in the case of the 10,000-ton ship.

Eleven-and-a-half knots is, of course, very much less than the economical speed of a 30,000-ton ship or even a 20,000-ton ship, the purpose of this illustration being solely to show the gain resulting from an increase in deadweight only.

SPEED OF TANKERS

As ships increase in size higher speeds are permissible. The economical average speeds for ships of 10,000, 20,000 and 30,000 tons deadweight are in the region of 11½, 13½ and 15 knots respectively, and taking the operating cost of the smallest ship as 100, the operating costs of the others are 64 for the 20,000-ton ship and 49 for the 30,000-ton ship. Thus by increasing the speed of the 20,000-ton ship to 13½ knots a gain of two points is made, and three points when the speed of the 30,000-ton ship is increased to 15 knots.

As regards the speed of ships it will be appreciated that for any given power the speed will differ slightly between loaded and ballasted voyages and also during bad and good weather conditions. The speeds mentioned throughout this paper refer to the average that can be expected for a period of one year when trading world-wide.

While a good case can be made for increasing the size of tankers, the reason for the speeds of some tankers recently constructed or at present under construction is not so apparent. One finds that tankers of the same deadweight have up to three knots difference in speed. If such ships are to carry the same kind of cargo and operate on similar routes the only inference can be that some will not operate at their most economical speed, since every ship of the same size and form has a definite economical speed which can be arrived at by calculation. Any speed above or below the economical speed means that under ordinary trading conditions the profit earning capacity of the ship is less than it ought to be.

In order to illustrate these statements the cost per ton of cargo carried by three 18,000-ton deadweight ships of the same hull form operating at 12, 13 and 14 knots respectively on a voyage U.K./Curaçao/U.K. will be given. The ships chosen have a length of 530 feet, a beam of 70 feet and a depth of 39 feet, the only difference being in the power of the propelling machinery.

Giving the operating cost of the 12-knot ship a basic figure of 100, the corresponding figure for the 13-knot ship is 98 and for the 14-knot ship 103. Thus it will be seen that the economical speed for this size of ship is in the region of 13 knots.

The ships considered in the foregoing examples are presumed to be propelled by steam machinery of an up-to-date type.

The foregoing calculations are for ships engaged in the Western Ocean trade, and do not apply to what might be termed special trades. For instance, where ships are operating on long, good weather voyages the economical speed will be found to be greater than for world-wide trading. Furthermore, where ships are engaged on short voyages it will be apparent that the port time, i.e., the time taken to load and discharge a cargo, will represent a much greater proportion of the time to complete the round voyage and introduces an entirely different set of circumstances which must be considered in determining the maximum overall economy.

The proposal to give tankers a speed which will enable them to sail in what were during the recent war considered fast convoys is, however, worth considering. But before reaching any conclusions regarding this proposal—and perhaps penalize ships during the whole of their useful life—we desire to know (a) will there be another war during the life of ships built now or in the near future, and (b) if there is to be another war and merchant shipping is again attacked, will the last war method of convoying be adopted in view of the greatly improved and vastly different weapons that may be used in future sea warfare? It may well be that the advent of the atomic bomb and the ability

to launch such missiles from a mobile source will render the convoy system not only obsolete, but positively hazardous.

The first of these questions is, presumably, unanswerable even by the most knowledgeable in the trend of world affairs, while, as regards the second question, it is unlikely that naval authorities will divulge before the war is upon us the form attacks on merchant ships will take, or advise shipowners what measures will be taken to defend their ships, apart from providing the defensive equipment used in the last war.

The only certain thing, whether ships sail in convoy or independently, is that the quicker they are able to complete a voyage the less likely they are to be attacked. If for instance, a ten-day voyage can be accomplished in nine days the chance of attack will have been proportionately reduced. It may be, of course, that the speed which enables the ship to do the voyage in nine days may bring it within danger and result in its undoing, whereas the slower ship would have kept out of range and arrive safely at its destination, but such circumstances cannot possibly be foreseen and in determining the safest speed for ships the assumption must be that the faster the ship the less likely is it to be attacked.

If we accept the view that the faster the ship the less likely it is to be attacked, the next question to be considered is "what is the speed that will render a ship immune from attack," if there is such a speed. Experience in the recent war with two particular ships tends to show that there is no such speed. These ships were specially designed during the war for conditions then existing and were given a speed of about 40 knots. The fact that the ships consumed a quantity of fuel greater than the quantity of cargo carried was unimportant in the circumstances, the chief requirement being to reduce to a minimum the risk of the small quantity of cargo carried not being delivered to its destination. The fact that one, if not both ships were attacked and sunk by the enemy with the weapons available at that time confirms that there is no practicable speed which will ensure immunity in the next war.

QUANTITY OF CARGO CARRIED BY FAST AND SLOW TANKERS

For the same length, breadth and depth a slow ship will carry more cargo per voyage than a relatively fast ship, but if several cargoes are carried over a period of, say, one year the fast ship will transport a greater quantity of cargo than the slow ship, within a certain range of speed.

As this fact is sometimes advanced in support of higher speeds it may be interesting to compare the fuel consumptions and quantities of cargo carried by two ships of the same overall dimensions, one having a speed of 13 knots and the other 16 knots. The approximate dimensions of the ships are, length 530 feet, breadth 70 feet and depth 39 feet. While the principal dimensions of the two ships are identical the block co-efficient, and incidentally the deadweight, of the faster ship is, of course, the smaller in accordance with well established rules in naval architecture.

The two ships are presumed to operate between Curacao and the U.K., and in one year the tons of fuel consumed by the propelling machinery and the tons of cargo carried in that time by the two ships are found to be:

	Fuel Consumed Tons	Cargo Carried Tons
13 knot ship	13,850	182,500
16 knot ship	20,500	186,000

The lines of the fast ship in the above example could, of course, be made fuller and result in a greater quantity of cargo being transported in the specified time, or vice versa, but to maintain the speed of 16 knots the power of the propelling engines would have to be increased, with a consequent increase in the quantity of fuel consumed over the same period. Thus it would be found that the extra freight earned would be more than offset by the cost of additional bunkers.

RESERVE POWER

An argument sometimes put forward in favor of ample power in a ship is that a ship so provided can on occasions avoid missing a tide when making port. Such a ship may once or twice a year save twelve hours' delay in entering port, but if feats of this kind are to be accomplished the reserve power must be of an order that will give the ship a speed increase of at least one knot. A smaller proportion of reserve power may by chance cause a ship to catch a tide that would be missed otherwise, but the circumstances from which such a benefit would accrue are so rare and unforeseeable that there is no sound basis on which to calculate.

The usual practice is to operate machinery in service at from 5 to 10 percent below the engine builders' rated power output in order to keep wear and tear and, consequently, maintenance costs, within reasonable limits. If, in addition to this, a ship is given sufficient reserve power to occasionally travel at one knot greater than the normal maximum speed, the machinery will, except on such rare occasions, be operating at not more than three-quarters the rated power. If the machinery is of the Diesel type this means that the fuel consumption will be 0.370 lb. per B.H.P. per hour instead of 0.357 lb., which in an installation rated at 5000 I.H.P. represents a difference of 0.46 tons per day. Assuming the number of days a ship spends at sea to be 320 per annum the cost to catch a tide on the few occasions when this becomes necessary is 147 tons per annum or £1300 at present-day Diesel fuel prices.

DESIGNING A TANKER

During the early days of the petroleum industry it was customary for the design and construction of a tanker to be left more or less to the shipbuilder, who, for his part, contracted to build a ship which would conform to the minimum regulations of a particular government or classification society and have a certain deadweight and speed. The speed was generally confirmed by sea trials carried out under favorable weather conditions, the propelling power installed being just sufficient to meet obligations. Consequently the hull form and power of machinery were designed to meet requirements under fair weather conditions at a minimum building cost, the question of the ship's performance in service not being of first importance to the shipbuilder.

All are aware of the great increase in the quantity and variety of petroleum products to be transported by sea. This has resulted in the need for improved design and construction. The tanker is now recognized as a ship requiring highly specialized knowledge in design and the best material and workmanship in construction, and many shipbuilders have even laid out their yards and added equipment to deal with this particular class of work. This, it will be appreciated, is necessary as, in addition to the difficult nature of some of the

products carried, it is not unusual for a tanker to be loaded at the rate of 2000 tons per hour with cargo at double the temperature of the ship and surrounding water. Present-day requirements also call for the ocean transport in bulk of cargoes such as bitumen (which may require to be heated to 250 degrees F., when the sea temperature is as low as 40 degrees F.), butane, propane and sulphuric acid, about the carriage of which the shipbuilder cannot be expected to have knowledge.

To the shipowner an efficient tanker is one which during its lifetime will meet particular requirements at the minimum cost per ton of cargo carried. To determine this, all costs involved must be taken into account which, in addition to capital charges, include such items as insurance coverage, wages, stores, victuals, bunkers, port charges and maintenance costs. It is in regard to the latter that we are mainly concerned here, since it will be appreciated that any failure in the structure of the ship or in the machinery, cargo pumping and other equipment will not only lead to large repair bills, but cause delay and consequent reduction in earning capacity. Moreover, there is the question of possible dislocation in the anticipated trading program to be considered.

It is in connection with such matters that the experience of the shipowners' technical advisers becomes invaluable. Additional stiffening at certain parts of the structure as found necessary to prevent fracture or allow for wastage, as well as modifications to the machinery installation and equipment that will provide safeguards against breakdown and off-hire periods, and although these generally involve additional initial capital expenditure it is regained many times over throughout the life of the ship when judiciously applied.

The main points in the hull design of a medium size tanker for world-wide service which affect economy in service are deadweight, average service speed and cargo handling equipment, and, while the first two have a definite interrelation, the latter is concerned with the length of time the ship must be in port—a very important factor in the earning capacity of a ship.

Other features in the design of a tanker which are mainly the concern of the shipowners' technical advisers are the sub-division of the cargo spaces, prevention of cargo admixture or contamination, safe handling of cargo, the disposal of dangerous gases and the layout of the cargo pumping arrangement.

The present-day demand for more and still more deadweight has in some cases tended to influence the ship designer towards increasing the ratio of length and/or breadth to depth in excess of what is considered good practice. The full effect of reducing the depth in relation to length and breadth may not at first glance be apparent.

A ship can be regarded as a long, narrow steel box, the load on which is represented by the lightweight of the ship plus the various items which make up the total deadweight, such as cargo, bunkers, fresh water, stores, etc. This weight is supported by the buoyancy resulting from displacement, which varies in degree from the ends to amidships, it being least at the ends and greatest amidships.

From the point of view of strength the smaller the depth of the box the greater will be the thickness and consequently the weight of steel necessary to obtain the required strength. Consequently the first point to be considered when inclined to reduce the depth without a corresponding reduction in length and breadth, is the question of additional steel weight, the amount of which will depend on the extent of deviation from normal proportions; the greater the deviation the greater will be the amount of extra steel required.

A matter of even greater importance is the question of deflection in heavy weather with the greater possibility of structural failure, for although the strength of the box of reduced depth may, by the addition of extra steel, be the same as one of normal proportions, the corresponding moment of inertia will be less. In this respect also the extent again depends upon the amount of deviation. Before adopting unusual proportions, therefore, it is essential that all such factors be given careful study.

It is not suggested that deflection as such—or to be more correct, flexibility—in a ship is undesirable. Actually all ships must be flexible to a certain extent. Theoretically the greater the flexibility the less should be the shock effect of heavy seas pounding the structure. Experience with riveted construction, however, has proved that too much flexibility results in slackening of rivets in the bottom and deck plating, which are the chief longitudinal strength members. In the case of all-welded ships the position is rather different as there are no rivets to become slack, but the possibility of fractures occurring owing to too much flexing must be taken into consideration.

The stresses set up in a ship during heavy weather under various conditions of loading and ballasting cannot be accurately determined by calculation. The standard of strength at present adopted by Classification Societies is the result of observations made over many years and represents what is considered necessary to provide a reasonable margin of safety.

Tankers in the region of 30,000 tons deadweight are, however, a comparatively recent innovation and the extent to which the present recognized relationship of length and breadth to depth can be exceeded with safety and economy in maintenance must remain a matter of opinion until more experience with such ships in service has been gained.

The improvements effected in tanker construction and operation during recent years are as follows:

(1) Improvement in accommodation and all matters which make for the comfort and convenience of officers and crew.

(2) The introduction of labor-saving appliances and mechanical and electrical aids to navigation.

(3) Larger and faster ships.

(4) The substitution of welding for riveting in hulls and superstructures and the employment of fabricated steel structures in place of large engine castings.

In the matter of accommodation it is now customary (in the Shell fleet it has been standard practice for some years) to provide each member of a ship's company with a separate room, while commodious recreation rooms containing libraries and broadcast receivers are provided for the officers, petty officers and crew. Hot and cold water are on tap in the elaborately equipped lavatories and provision is made in these spaces for washing clothes, to the extent of providing electrically operated washing machines. It is also now common practice to ventilate mechanically all spaces in the accommodation and provision stores.

Regarding labor-saving appliances, it is no longer necessary for cargo tanks which have previously carried viscous cargoes to be cleaned out by hand, the work now being done by mechanical means and the only labor required being to remove the washed sediment from the bottoms of the tanks. Also, the cargo spaces of tankers which have previously carried volatile cargoes can now be gas-freed by means of greatly improved gas ejectors instead of by windsails, which entailed labor and constant

attention because of the changing direction of wind. Moreover, it is common practice to supply tankers with pneumatic chipping hammers for quick and easy removal of scale in spaces not likely to contain explosive gases, while the upper structures are painted by compressed air sprayers.

WELDED CONSTRUCTION

The most notable development in ship construction during the present century is undoubtedly the method of joining together the various plates and angles which go to make up a ship's main structure.

The chief advantage of substituting welded for riveted construction is the substantial saving in weight of material necessary to construct a ship of a given size. This saving is mainly due to the elimination of plate overlaps and rivet heads.

A further advantage of welded construction is the elimination of projecting plate edges, which when vertically disposed tend to retard the forward movement of a ship and increase slightly the power required to give a ship of certain dimensions a given speed. Projecting plate edges which are horizontally disposed do not impede forward movement, but they have an anti-rolling effect, so that an all-welded ship might be expected to roll to a greater extent than a riveted ship.

It may take a longer time to weld than to rivet a seam of a given length; also, the plate edges require more attention before welding. If, however, the time required to drill and fair holes of a riveted seam is taken into account there is little difference in the time taken to weld or rivet a given length of seam. The actual figures are: (a) 84 man-hours to prepare and weld a seam 100 ft. long between plates 1-in. thick, and (b) 75½ hours to prepare and rivet a similar seam. The figure given in (a) is for manual work. If the seam is machine welded, a practice that is being increasingly adopted, the number of man-hours is 39, or little more than half the time required to rivet.

There is no difference in the relative strengths of welded and riveted seams. While greater care is necessary for welded construction, and although a badly riveted seam is less likely to cause disaster than a badly welded seam, there is no doubt that welding will soon become universal practice in ship construction, particularly as the efficiency of a weld can be ascertained by X-ray apparatus which has been developed for such purposes.

The great weight advantage in making large parts such as bedplates, frames and gearcases of marine engines of welded steel sections in place of iron or steel castings is not as yet generally recognized, but the day is not far distant when large castings for such purposes will be entirely superseded. Another advantage is that the loss in time and money which accrues when a defective casting is not discovered until the machining stage is well advanced, is eliminated. Furthermore, repairs are more easily effected in a welded steel structure in the event of damage.

Rigidity is an essential quality in the engine parts referred to, and the fear that a fabricated steel structure, such as a bedplate, is less rigid than a casting has deterred many from taking advantage of this recent development. The Shell Group, however, have had such structures in service for many years without experiencing trouble.

THE MOST EFFICIENT DIESEL ENGINE

E. C. MAGDEBURGER

THE AUTHOR

devoted a lifetime to the development of the diesel engine for the U. S. Navy. He was born and graduated in mechanical engineering in old Russia. Fascinated by Rudolf Diesel's concept of the "rational heat motor" while building gas engines in Germany he began his diesel career with a Russian licensee of MAN. In 1910 he came to this country and built large gas engines for Allis Chalmers Company and crankcase compression oil engines for Fairbanks-Morse Company. In 1912 he joined Busch Sulzer where a line of engines under Sulzer license was developed and submarine engines were built to power Lake submarines. Since January, 1922, he has been with the Bureau of Ships or its predecessor. Many of his past contributions to the subject were published in the JOURNAL.

Thermal efficiency of any engine is a commonly accepted yardstick for measuring the part of the thermal energy of fuel which the engine can convert into mechanical torque of its rotating shaft. To the U.S. Navy, operating as it does over great distances, high thermal efficiency of its propelling plants is particularly important, because it results in greater cruising radii on a given amount of fuel or less weight of fuel to be carried for any desired cruising radius. To a merchant ship it simply means reduction of operating expense and better competitive position.

But there is one other aspect of high thermal efficiency that must not be overlooked especially in the Navy where the economic disadvantage of higher fuel consumption is easily accepted in exchange for such advantages as simplicity and greater specific output. The greater the part of the fuel which can be converted into useful mechanical torque the less is the wasted thermal energy which plagues engine parts exposed to high temperatures by overheating them. Wasted fuel also makes smoke and produces deposits within the combustion chamber. All of these limit the maximum useful output of the conventional diesel engine and increase its first cost and the cost of maintenance in service.

The original diesel or compression ignition engine maintained its leadership as the most efficient available source of power throughout its brilliant career to date. It won the war for us—over 50 million horsepower in diesel engines were called upon to propel a wide variety of naval craft burning a minimum of safely handled fuel.

There were at least two major steps in its development substantially improving its thermal efficiency—(a) The adoption of mechanical injection of fuel, through elimination of the parasitic load of the injection air compressor and (b) the introduction of turbocharging which

converted the heretofore wasted energy of the exhaust gases into more charging air, i.e., of higher than atmospheric pressure. The recently completed development of a new highly supercharged MAN engine has demonstrated how the thermal efficiency of the diesel engine can be further improved substantially.

Five years ago, facing a grave national problem of resurrection from the devastation of the recently ended war, MAN decided to resume the development of supercharging pursued during the war on four cycle submarine engines (see JOURNAL A.S.N.E. February 1948, p. 13) as the most promising means to regain leadership. A three-cylinder engine of 11.8″ bore and 17.7″ stroke was available. To supply the charging air to this test engine another engine was used to drive a Roots blower. The problem was to match the turbine power that could be produced from the exhaust gases of the test engine with the power required to produce this air charging it, as it would have to be in a turbocharger.

Buchi in his original turbocharging system had to be satisfied with 50 per cent increase in the engine output while supercharging with air of 5-6 psig. He used the excess air as delivered by the blower of the turbocharger to dilute the engine exhaust gases and to reduce their temperatures to levels tolerable to the turbine. This was accomplished by introducing scavenging or blowing out the cylinder contents when the piston is near the upper dead center through the overlapping valves which action, incidentally, cooled the cylinder combustion space also. The 100 per cent turbocharging or doubling of the original output of the atmospheric aspirated engine now used on more modern types of turbocharged engines is based on 15-16 psig charging pressure. With more air in the cylinders more fuel is burned and more energy becomes available in the exhaust gases to drive the turbine. Scavenging and valve overlap are retained. However, to reduce exhaust gas temperature to the same level of tolerance by turbine blades it was necessary to introduce water cooling of the charging air. And that is the key to success with this higher level of supercharging.

Miller at Nordberg cools by expansion. The charging air is compressed to a higher pressure, and therefore only a reduced amount of it can be handled by the blower of the turbocharger. Then only a part of the cylinder volume is charged with it, say the first 80 per cent of the stroke. During the remaining part of the stroke it is expanded to a lower initial temperature for the subsequent compression. Obviously the maximum possible cylinder output is thereby reduced and the thermal efficiency penalized by the waste of exhaust gas energy for cooling.

From the basic equations of compression we know that air compressed to 1/13 of its original volume will be under 31.9 times higher pressure and of 2.45 times higher temperature in absolute units.

$$\text{For } n = 1.35, \frac{V_1}{V_2} = 13, \frac{P_2}{P_1} = 13^{1.35} = 31.9, \frac{t_2}{t_1} = 13^{.35} = 2.45$$

It is therefore obviously more economical to cool the air before it enters the compressor by t° than to cool the same original volume of air after it is compressed by 2.45t° to get the same result.

$$\text{For } t'_1 = t_1 - t \qquad t'_2 = 2.45\,(t_1 - t) = t_2 - 2.45t$$

Furthermore, the original volume will hold a greater weight of air of lower temperature and therefore will be capable of developing greater BMEP by combustion. And instead of wasting a large quantity of relatively hot charging air for lowering the temperature of combustion gases by scavenging of the combustion space this air can be used for combustion with conventional excess of air when it is cooled, before entering the working cylinder, in a conventional tubular cooler by circulating water.

The only limit to cooling the charging air is that the reliability of ignition by compression must not be jeoparized, especially during starting.

MAN used a compression ratio of 13 to 1 and under overload conditions the charging pressure would reach 3 atmospheres absolute (27.9 psig) and the compression pressure $3 \times 31.9 = 95.7$ atm. = 1360 psia. Then, while the peak pressure would rise to about 2000 psi, the temperature of the exhaust gases did not exceed 550°C = 1020°F to the turbine, with considerably lower temperatures of exhaust of individual cylinders, so that uncooled cast aluminum pistons were used with excellent results, MAN has taken the position that high pressures can be easily met by design modifications, while higher temperatures require materials not yet available.

The new engine established beyond doubt, the maximum charging pressure it is possible to obtain without excessive gas temperatures and also proved that the well known previously observed improvement in thermal efficiency with increasing charging pressures continued to this maximum charging pressure.

To MAN belongs the credit for having eliminated valve overlap and scavenging and the use of charging air for dilution of hot combustion gases and substituting instead a more extensive cooling of charging air by circulating water.

The three-cylinder engine tests made in 1949 demonstrated that with suitable modifications it would be possible to supply the cylinders with as much as three times the weight of atmospheric air, i.e., raise the charging air pressure to 27.9 psig or 3 atmospheres absolute. The original problem was thereby reduced to designing a most efficient turbocharger to convert the energy in the exhaust gases into maximum weight or maximum pressure of charging air.

Encouraged by these test results MAN decided to enlarge the original engine to six cylinders and to build a turbocharger to suit this engine. A five-stage reaction turbine for greater efficiency was chosen to drive a nine-stage axial blower with the added final stage being centrifugal to avoid too short blading. To reduce the horsepower required by the blower, intercooling between the axial and centrifugal blowers was incorporated as well as aftercooling to reduce the temperature of the charging air delivered to the engine. Valve timing of the normal atmospherically aspirated engine with only nominal overlap was retained in the new engine to eliminate waste of charging air and compression ratio was reduced to 13. Three different configurations of exhaust piping to the turbine were designed and tested in an effort to increase the turbine output with consequent rise of charging air pressure in order to utilize the blowdown energy of the individual cylinder exhausts.

In other respects, however, the engine does not differ from a conventional diesel engine, except that it has larger bearings to cope with higher cylinder pressures. The fact that it has crossheads is explained by the MAN desire to reduce lube oil consumption as well as fuel consumption, a feature of considerable importance under present circumstances in Germany. Two intake and two exhaust valves and a central fuel valve with larger size individual fuel pumps driven by the camshaft con-

stitute the rest of the design features of importance. The timing of fuel injection was chosen to secure combustion of part of the fuel charge at constant volume in dead center with the peak pressure, however, not to exceed 1800-2000 psig.

In the Spring of 1951 this new highly turbocharged engine was tested by such independent authorities as professors Walter Pflaum of Berlin Technical University and G. Eichelberg of Zurich Federal Technical University who are scheduled to present their joint report at the Annual Meeting of the VDI (Society of German Engineers) in December 1951. It has been my privilege to discuss this joint report with both of them and it is hoped that it will be published in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS in the near future.

MAN is willing and anxious to demonstrate the new engine in operation and to prove the claims made for it. A check test was witnessed with BMEP of 240 psi for continuous operation of 24 hours equivalent to 1400 BHP at 400 rpm, followed immediately by 2 hours at a BMEP of 284 psi developing 1600 BHP at 380 rpm.

Engine Service	*New MAN Test*	*GM Model 258S Naval*	*FM Naval*	*Nordberg Gas Diesel Alcoa at Lavacas, Texas*
Cycle-No. Cylinders	4-6 in line	4-V-16	2-10 O.P.	2-11 radial
Bore and Stroke	11.8 × 17.7	9½ × 12	8⅜ × 10	14 × 16
BHP at rpm	1400 at 400	1800 at 900	1600 at 720	1650 at 360
BMEP-psi	237.7	116.38	84.9	67.0
BHP at rpm	1600 at 380	2000 at 900	2000 at 800	1800 at 400
BMEP-psi	286.0	129.2	95.6	65.8
Fuel Consumption at rated load-lb/BHP hr.	.308	.38	.38	...

In the above table are assembled data of three other engines for easy comparison with this new MAN engine—a turbocharged General Motors four cycle engine, Model 258S, a Fairbanks-Morse opposed piston two cycle both of which were used in large numbers during the war in the Navy, and a Nordberg radial two cycle engine operating on gas as well as fuel oil, 120 of which were installed in the Alcoa plant at Lavacas, Texas—claimed to be the largest in the world (200,000 BHP). The tabulated figures demonstrate convincingly just how much of a step ahead is represented by this new MAN engine both as to specific load carrying capacity of the cylinder BMEP and the thermal efficiency or specific fuel consumption in lbs/BHP hr of the engine.

The check test referred to above fully confirmed the claims made for the engine and MAN engineers see no reason why these new characteristics can not be grafted on to a high speed naval type of engine, which will add light weight and compactness due to higher rpm to the high thermal efficiency and load carrying capacity demonstrated.

The objective of this brief report is to show that no magic is required to explain the results obtained by MAN. They were thermodynamically predictable and awaited only the pressure of necessity to generate the courage to make them measurable. It is up to the Navy now to make use of the new engine ahead of the rest of the world.

COMFORT AIR CONDITIONING—PASSENGER SHIPS

ACKNOWLEDGMENT

This article is reprinted from "Refrigerating Data Book Applications," Third Edition, 1950, published by the American Society of Refrigerating Engineers. The author is S. W. BROWN, Chief Engineer Marine Department, Carrier Corporation. Mr. Brown is a member of the American Society of Naval Engineers.

The present day demand for travel comfort, plus the increased competition from the air lines, makes air conditioning mandatory for the modern passenger ship. All post-war constructed passenger ships are extensively air conditioned. Passenger ships constructed prior to the war have added air conditioning to many of their spaces upon reconversion.

Whereas air conditioning was previously fitted into a ship after all other machinery and services were assigned, today it is generally recognized by the Naval Architect and Marine Engineer that consideration should be given to the air conditioning plant in the initial stages of design.

The following are some of the spaces provided with air conditioning on modern passenger liners: passenger staterooms, officers' and crews' quarters, dining rooms, officers' and crews' mess, smoking rooms, dance pavilion, theatre, game room, lounge, library, hospital ward, operating room, doctor's office, purser's office, stewards' office, beauty parlor, barber shop, print shop, and children's playroom.

The design of air conditioning systems for passenger ships present problems far more serious than usually encountered ashore. The compactness of ship accommodations, combined with the complex structure of a ship, involves many problems which affect not only the selection of the equipment, but also the design. The subdivision of the hull into a number of watertight compartments necessitates the employment of a multiple number of fan systems in order to avoid the penetration of these watertight bulkheads wherever possible.

GENERAL CONSIDERATIONS

Factors to be considered in the design of an air conditioning plant for shipboard are as follows:

I. The plant should be designed to function properly under conditions of roll and pitch to which a ship is normally subjected.

II. The materials of construction should be suitable to withstand the corrosive effects of sea air and sea water.

III. The system should be designed for uninterrupted operation during the voyage. Since ships en route cannot be easily serviced, even from land based

planes, some standby capacity, spare parts of all essential items and extra refrigerant charges should be carried.

IV. The system should be designed so that there will be no objectionable noises or vibration. Although the ship's engines and auxiliaries may veil the noise and vibration of the air conditioning system while in voyage, it must be realized that the engines and auxiliaries are shut down while the ship is in port and that the noise level under port conditions must be within the acceptable limits established for land practice.

V. In view of the high premium for space on shipboard, the equipment should be designed to occupy a minimum of space commensurate with cost and reliability. Weight should be held to a minimum.

AIR DISTRIBUTION

The problem of providing draftless air distribution on shipboard requires careful study, because of the relatively low headroom and other construction requirements of a modern ship, such as, projecting beams, indirect lighting coves and dome-shaped ceilings. Draftless air distribution is even more difficult to achieve in staterooms, particularly where pullman berths are provided. The methods of air distribution fall into three general classifications: (1) The side wall type which includes bar-type grilles and half-round diffusers; (2) The upward discharge type in which a bar-type grille is generally used. This method is confined to floor mounted installations, such as, induction units; (3) The ceiling type which consists of round or half-round, square or rectangular, flush or projecting diffusers. Another method of ceiling type distribution is the perforated ceiling panel.

DESIGN CONDITIONS

In determining the proper outside design temperatures, consideration should be given to the intended service of the vessel. In addition to their regular itinerary, many liners frequently make off season cruises during which more severe heating and cooling loads may be encountered. The selection of the ambient design should be based upon the temperatures incurred in port rather than upon the temperatures prevalent during the voyage. In general, for the cooling cycle, outside design conditions for North Atlantic runs are 95 F dry bulb and 78 F wet bulb; for semi-tropical runs 95 F dry bulb and 80 F wet bulb, and for tropical runs 95 F dry bulb and 82 F wet bulb. For the heating cycle, 0 F is usually used as the design condition unless it is definitely known that the vessel will always operate in higher temperature climates.

Effective temperatures from 73 to 75 F are usually selected as inside design conditions for cooling cycle. Public spaces are often designed for 80 F dry bulb and 55% relative humidity. Passenger staterooms are designed for 80° dry bulb with a relative humidity of 50% or less.

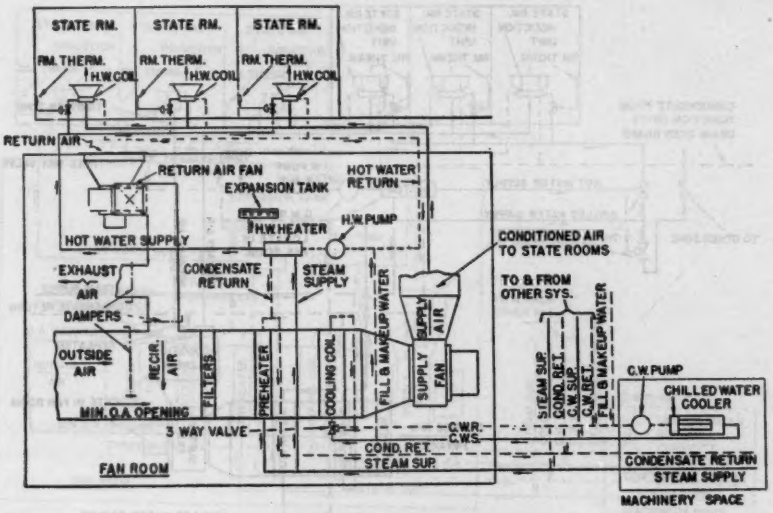

Fig. 1. Type "D" System.

TYPICAL AIR CONDITIONING SYSTEMS

Comfort air conditioning systems installed on shipboard are classified as follows:

A. Those serving passenger staterooms

B. Those serving crew's quarters and other similar small spaces

C. Those serving public spaces.

Various types of systems have been applied to date for passenger staterooms and crew's quarters. These are identified as follows:

1. Terminal reheat system which is commonly known as the USMC class "D" system. A diagrammatic of this system is shown in Fig. 1. In this system, conditioned air is supplied to each space in accordance with its maximum design load requirements. The room dry bulb temperature is controlled by means of reheat in which a room thermostat automatically controls the volume of warm water passing through the reheat coil which is provided for each controlled space. In this system a mixture of fresh and recirculated air is circulated through the ductwork to the conditioned spaces. A minimum of fresh outside air is mixed with return or recirculated air in a central station system where it is filtered, dehumidified and cooled by the chilled water cooling coils, and distributed by the supply fan through conventional ductwork to the spaces treated. Automatic dampers are frequently provided to increase the outside air to a maximum of 100% as the load falls off. In accordance with usual practice, no recirculated air is permitted for operating rooms and hospital spaces. When 100% outside air only is used, the system is designated as Class "D1." When heating is required, the conditioned air is heated at the central station to a predetermined temperature. In addition, hot water is circulated to the reheat coil to provide additional heating capacity, as required to maintain the desired room temperature.

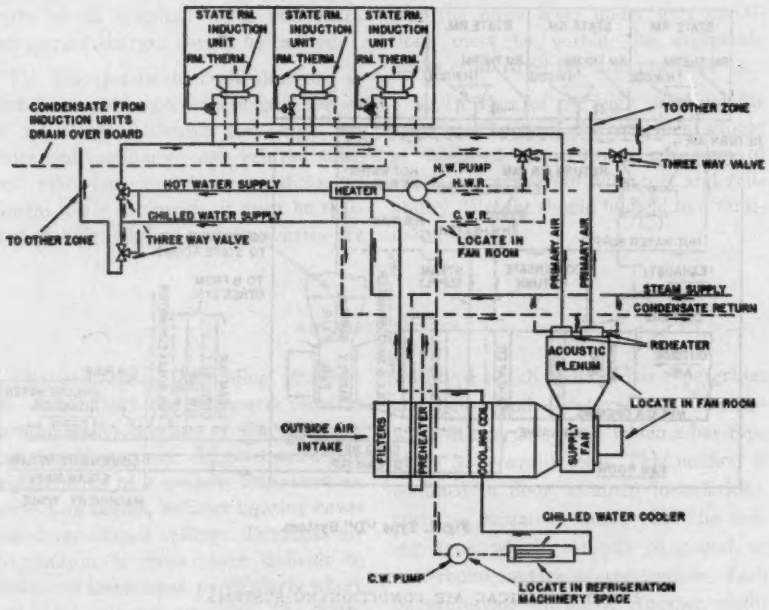

Fig. 2. Type "E" System.

2. A second type of system for passenger staterooms and other small spaces is the Conduit Weathermaster system which is designated as Class "E" system by the USMC. In this system, shown in Fig. 2, a central station is provided for conditioning the primary air only. The primary air is circulated to induction units located in each of the spaces to be conditioned. The induction nozzle, through which the primary air passes, induces a fixed ration of room air to flow over a water coil and mix with the primary air supplied to the unit. The mixture of fresh air is then discharged to the room through the supply grille. The room air passing across the water coil is either heated or cooled, as required. The flow of water to the water coil can be controlled either manually or automatically to maintain the desired room conditions. In this system, no return or recirculated air ducts are required, since only a fixed amount of fresh air need be conditioned at the central station equipment. This small amount of conditioned air, which is cooled to a relatively low dew point, is circulated at high velocity and at high pressure through small ductwork, thus reducing very substantially the space required for the air distribution.

When heating is required, the primary air is heated in the central system to a predetermined temperature. In addition, hot water is circulated to the induction units to provide additional heating capacity, as required to maintain the desired room temperature.

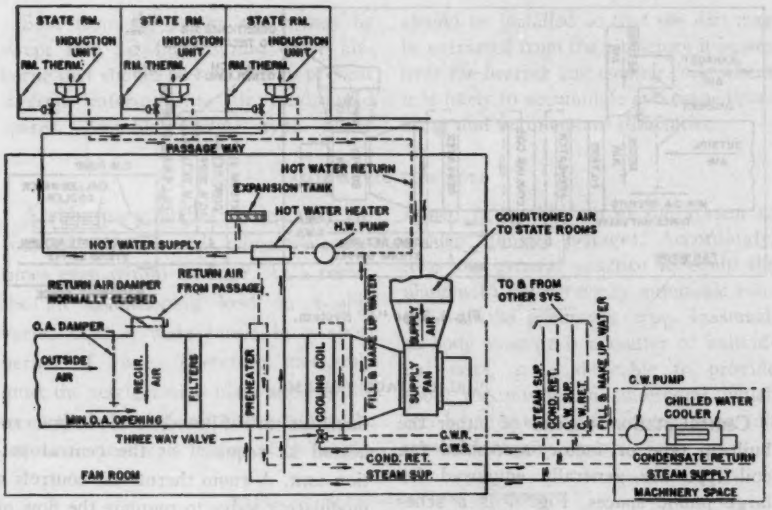

Fig. 3. Reheat System with Induction Units.

3. A third type of system used for small individual spaces employs reheat with induction units (Fig. 3). The primary air, which is filtered, cooled and dehumidified to a relatively low dew point in the central station system, is distributed through a duct system at moderately high velocities and pressures to the staterooms or individual spaces which are equipped with induction units. The primary air, in passing through the nozzle in the induction unit, induces room air to pass over a hot water reheat coil. A room thermostat automatically controls the flow of hot water to the heating coils to maintain the desired room temperature. In this system, as in the terminal reheat system, the conditioned air must be cooled sufficiently to maintain the required room conditions under the maximum design load. Therefore, reheat is required when the load is less than the maximum design. With this system, the primary air may be cooled to a relatively low temperature, since it is mixed with warmer room air at the induction unit before the mixture is discharged into the room. Accordingly, the quantity of air circulated through the ductwork can be substantially reduced, resulting in the use of smaller duct sizes. When heating is required, the primary air is tempered in the central station system to a predetermined temperature. In addition, hot water is circulated to the induction units to provide additional heating capacity as may be required to maintain the desired room temperature.

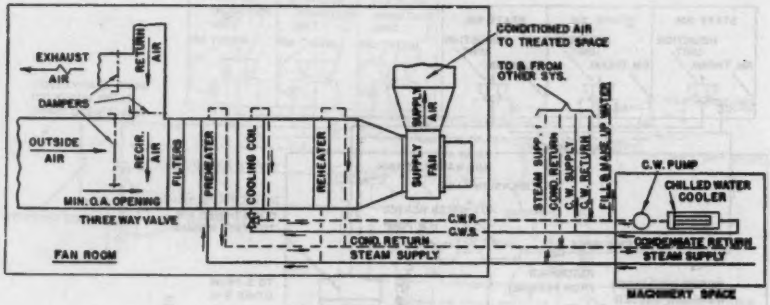

Fig. 4. Type "A" System.

PUBLIC SPACE SYSTEMS

Central station systems of either the built-up type or factory-assembled fan coil type are generally employed for large public spaces. Fig. 4 is a schematic diagram of a typical public space system classified by the USMC as a type "A" system. (Under certain conditions, it is found desirable to use all outside air and avoid the use of return ducts. Where all outside air is used, the system is designated as a USMC type "A1" system.) Outside air and return air are filtered, cooled, and reheated as required at the central station unit. A room thermostat controls a modulating valve to regulate the flow of steam to the reheat coil and maintain the desired room temperature. During the heating cycle, the dampers are arranged to admit 100% outside air which is filtered and preheated at the central station apparatus and reheated as required by the room thermostat.

VENTILATION REQUIREMENTS

Ventilation requirements for passenger vessels generally exceed acceptable standards for land practice. Copious amounts of fresh air are desirable to eliminate odors from the bilges, engine room, from fueling and from cargo.

In public spaces, a minimum of 12.5 cfm, per person, of fresh air is provided. The quantity of fresh air is automatically increased to 100% on light loads. On the Class "D" system, a minimum of 12.5 cfm, per person of fresh air is supplied. Here again, the outside air is increased to a maximum as the load decreases. All outside air is used on the heating cycle. However, it is customary to reduce the fan speed to circulate only 50% of the air. With the reheat type induction unit system, which can be designed for either all outside air or a mixture of recirculated and outside air, a minimum of 12.5 cfm, per person, or a minimum of a 30 minute air change, whichever is the greater, should be provided. When the system is designed to recirculate a portion of the air, automatic controls are arranged to increase the quantity of outside air as the load decreases. A minimum of 25 cfm, per person, of fresh air should be supplied for the conduit induction unit system.

FILTERS

Soot from the stacks which may be swept into air intakes and other airborne dirt should be removed to prevent it from entering into air conditioned spaces. Suitable marine type filters should be installed so that the dirt may be extracted from the air before it passes over the heating and cooling coils where it is likely to accumulate and cause operating and maintenance difficulties.

AUTOMATIC CONTROLS

A ship, on a single voyage, may encounter extremes of climates—sometimes even within an hour. As a result, the air conditioning load on a ship varies over a wide range in a short period of time. Therefore, not only must the refrigeration plant successfully meet these variations in load, but the automatic controls must also be designed to readily adjust the system to sudden climatic changes. Accordingly, it is the general practice to equip the plant with the necessary automatic controls of the pneumatic type. Inasmuch as body comfort is a matter of individual taste, it is desirable to provide room thermostats in staterooms which can be adjusted by the occupant to suit personal requirements.

TYPES OF REFRIGERATION SYSTEMS

Refrigeration machinery generally used on passenger vessels falls into two categories: (1) Freon-12 reciprocating compressors for capacities under 100 tons; (2) Centrifugal compressors for capacities in excess of 100 tons.

Some combination passenger and cargo ships fitted with considerable refrigerated spaces have used multiple duplicate compressors in which one of the compressors serves as a standby for the refrigeration as well as the air conditioning system.

Steam jet refrigeration has been used on foreign passenger liners, but has never been applied to any American passenger vessel. A new type of absorption machine designed for chilled water service, utilizing either low pressure or high pressure steam with lithium bromide as the absorbent and water as the refrigerant, offers many interesting possibilities for shipboard application.

REGULATORY BODIES

Vessels constructed in U.S. yards which are to operate under the U.S. flag come under the jurisdiction of the U.S. Coast Guard. Accordingly, the air conditioning equipment must conform to the Marine Engineering Rules and Marine Standards of the U.S. Coast Guard. The design of the equipment and the installation must also comply with the requirements of the U.S. Public Health Service. This involves principally the rat-proofing of all openings into the ventilating system. Although air conditioning installations do not ordinarily come under the cognizance of the American Bureau of Shipping, equipment should be manufactured wherever possible to comply with the American Bureau of Shipping Rules and Regulations. ASRE Standard 26, "Recommended Practice for Mechanical Refrigeration Installation on Shipboard," provides a guide for the installation of air conditioning equipment.

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SHOTPEENING AS A FACTOR IN THE DESIGN OF GEARS

ACKNOWLEDGMENT

The July, 1951, issue of "Mechanical Engineering" contained this article by JOHN C. STRAUB, of the American Wheelabrator and Equipment Corporation, Mishawaka, Indiana. It was presented at the Spring meeting of the American Society of Mechanical Engineers in Atlanta, Georgia, April 2-5, 1951.

INTRODUCTION

When a pair of gears is required to operate at high loads or high speeds or both, particularly in cases where weight and size are at a premium, it becomes important to consider shotpeening in the design. The greater the required horsepower per pound of transmission equipment, the more vital becomes the design of the gears which will be called upon to do the job. There is no intention of minimizing the importance of the other factors which go to make up a set of gears with high load-carrying capacity. Any significant shortcoming in forming the gear blanks, machining the teeth, control of material, heat-treatment, and the like, can nullify completely the advantages of an excellent design. Any confidence which the gear designer may have in the satisfactory operation of the gears which he has designed presupposes good manufacturing practice. This paper is concerned primarily with the design of the gear teeth themselves, particularly on spur and helical gears.

TYPES OF FAILURE IN GEAR TEETH

In general, gear-tooth failures may be grouped into three classifications:

1. Tooth breakage; this type of failure, in which an entire tooth or large portion of a tooth is broken out, is due to bending stress and is commonly known as a "fatigue" type of failure.

2. Pitting; this type of failure is characterized by pits or small craters in the contacting surface of the tooth. It is usually found on the pinion tooth at or somewhat below the pitch line. This is also a fatigue type of failure.

3. Scoring; this type of failure, sometimes referred to as spalling, scuffing, or galling, is distinctly different from the first two mentioned and is evidenced by a decided roughness on the working-tooth flank as though the mating surfaces had seized. As a matter of fact, it is generally accepted that scoring is actually the result of welding the two surfaces together and then tearing them apart. It is caused by high compressive stress, in combination with high sliding velocity of the tooth surfaces upon each other. Whereas bending and pitting failures are most likely to occur after a considerable amount of service, scoring is most likely to occur in the early stages of operation, if at all.

A pair of gears may be subject to any one or all of the foregoing types of fail-

ure, depending upon the type of service. In some cases this may result in a compromise between bending strength on one hand and scoring resistance on the other, because in some respects a design factor may favor one at the expense of the other. However, in highly stressed gears, scoring resistance may be increased by proper design consideration which utilizes greater bending strength.

It should be mentioned that in highly stressed gears, that is, gears which are required to transmit high horsepower with minimum weight, a good design is not necessarily obtained by even the most careful study of the problem from the standpoint of pure mechanics and strength of materials. In a simple beam, in which the exact magnitude and location of the load are known, it is a relatively simple matter to determine such factors as the maximum deflection, the bending moment, and nominal stress within reasonable accuracy.

When dealing with a pair of gears, however, we have an entirely different situation. It is true the geometry of gear teeth has been developed carefully and accurately, and, where necessary, gears are made with a high degree of precision. But when a pair of gears is mounted in a housing and the service load applied to the shaft, it is far from an easy problem to determine with any degree of accuracy the distribution of the load on the teeth, even though the laws of mechanics are considered most carefully, unless we have at our disposal sufficient test data to indicate what the distribution of load on the teeth is likely to be.

BENDING STRENGTH

In computing the bending strength of a pair of gears, one of the first questions that arises concerns the load on the teeth. Since a mathematical analysis does not yield a direct answer, another possibility is to assume that the transmitted load is distributed on the teeth in a somewhat logical manner. But there are several ways in which the load might be distributed on the teeth. For example, it might be assumed that the teeth are perfectly formed, perfectly spaced, and perfectly concentric. It may be further assumed that the deflections of the housing, bearings, and shafts are perfectly symmetrical. If these assumptions are correct, it would follow logically that the maximum bending stress in a tooth occurs at that point in the tooth action at which one tooth carries the entire transmitted load at the highest point on its profile, or the greatest distance from the root. Since we assumed in the beginning that the gears are perfectly symmetrical with respect to each other, it will follow logically that if these conditions exist even while the gears are transmitting load, the load carried by one tooth is distributed uniformly over the entire length of that tooth.

On the basis of the foregoing assumption, the problem appears to be well defined and not at all difficult. However, we must remember that this entire line of reasoning started out with a number of assumptions. It would have been just as logical had we assumed that the teeth are not perfectly formed, that they are not perfectly spaced, that they are not perfectly concentric, or that the deflections are not such that the mating gears are perfectly aligned while transmitting the load. Any one of these deviations from the original set of assumptions is likely to offset seriously the accuracy of our conclusion as to how the load is carried on the teeth.

How then are we to know the distribution of the load, and, therefore, to compute the load-carrying capacity of the teeth? One way would be to make

extensive measurements and computations on actual sets of gears under load. But this is a tedious and expensive procedure even for one set of gears, and, when the job is finished, it tells us nothing of the conditions which may exist in another design, or for that matter even in the same design mounted in a different housing.

Another way would be actually to test a large number of gears of different designs and different load-carrying capacities under conditions as nearly as possible like those obtained in actual service for which the gears have been designed. With complete data on the test procedure, cycles to failure, and design information, provided there are sufficient test data, the results can be analyzed statistically. This latter is the one which was used in selecting the method of bending-stress calculation involved in this discussion.

The procedure used in this selection was as follows: Data were obtained direct from different manufacturers on a large number of dynamometer fatigue tests on spur and helical automotive-type transmission gears. Complete information was obtained on the design and test results of each pair of gears.

A number of methods of stress computation were then set up, each on the basis of a different set of assumptions. In order to determine which, if any, of these methods would give consistent results, a "stress" value was computed for every gear included in the tests. The stress values obtained by each method were plotted on a log-log chart against the average number of cycles at which fatigue failure occurred.

As we would expect, some of the methods of stress calculation were immediately disqualified by the fact that no consistent relationship was thus obtained between the stress, as calculated, and the fatigue life from the test data. The results as calculated from other methods, however, did show a definite relationship between the calculated stress values and the average fatigue life from the test data, and it was not a difficult matter to select that method which showed the most consistent relationship.

There is no intention to imply that the method selected, or any other method of stress computation, based on the gear design, is capable of predicting exactly the life of a pair of gears under a given load. Anyone who is familiar with the characteristics of fatigue failure will realize that supposedly identical units of a given machine part, tested under identical conditions, are subject to a considerable variation in fatigue life. Such variations are inevitable in fatigue failure, regardless of the shape or accuracy of the parts involved. However, this inevitable variation in fatigue life need not be a source of discouragement, because by using the average results from a variety of designs, the influence of the design factors can be determined with reasonable accuracy.

Finally, the method of calculation which was selected, according to the procedure outlined, was based on the assumption that the load is distributed uniformly on the average total length of contact lines. The tooth-strength factor, commonly known as the "X-factor," is obtained from a layout in the section normal to the tooth, with the load applied at the tip of the tooth in the same manner as originally described by Wilfred Lewis. A complete description of this method of calculating bending strength was published some years ago (1).*

An appraisal of the effectiveness of this method can be obtained from the chart in Fig. 1 (reference 1), which shows the relationship between the stress number as calculated and the average life of the gears as determined

* Numbers in parentheses refer to the Bibliography at the end of the paper.

by the dynamometer tests. Included in the chart are 155 tests on a variety of designs.

All of the tests, represented in Fig. 1, were made on gears of approximately 60 RC surface hardness, with no surface treatment other than the heat-treatment itself.

By the use of this method, the design of a proposed pair of gears can be analyzed, and in some cases an appreciable gain in fatigue strength can be obtained by changes in the tooth proportions. For example, the requirements might permit a decrease in the diametrical pitch or an increase in pressure angle, either of which would decrease the bending stress.

It is quite evident, of course, that any strength calculation cannot be expected to overcome poor manufacturing practice. There are numerous factors in processing which can affect the life of the gears seriously. One such factor is the nature of the tooth fillet, with regard to stress concentration resulting from deep toolmarks. Deep grooves in the tooth fillet can be more serious than the stress concentration due to the fillet itself. To minimize this condition it is good practice to use a chamfer, or better still, a rounded corner on the hob or cutter.

Since the method of calculation is based on average test results, a particularly good manufacturing procedure is likely to give better results than indicated by the average line.

The validity of a method of calculation of bending strength on automobile spiral-bevel-gear teeth was established some years ago (2) on the same basis as that described in the foregoing for spur and helical gears. It is interesting to note that these two methods differ sharply with regard to the assumed load distribution as well as the section in which the layout is made. Each method agrees very well with test results of gears of the type in question, but neither is valid for the other type.

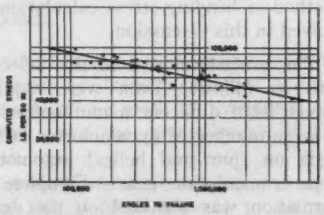

FIG. 1 FATIGUE CHART OF 155 HELICAL AUTOMOBILE TRANSMISSION GEARS USING PREFERRED METHOD OF CALCULATING TOOTH STRESS

SHOTPEENING FOR INCREASED STRENGTH

In gears which are intended to carry high stresses, a pronounced increase in fatigue strength can be obtained by shotpeening (3, 4, 5) the teeth. This is a process which is becoming more widely used as its advantages become more fully understood.

It is an established fact that shotpeening is very effective as a means of increasing fatigue strength. Its effectiveness in overcoming fatigue failures in a given design of a machine part is well known. But its advantages in designing for greater fatigue strength and greater utilization of material are often overlooked. The increase in allowable stress for a given fatigue-life requirement will vary with that life requirement, that is, the greater the required life, the greater will be the benefit derived from shotpeening. However, even under severe requirements, an increase of 10 percent in allowable stress is conservative.

Fig. 2 shows a fatigue line similar to that in Fig. 1, based on the same method of calculation, but from considerably more data than those originally accumulated. The lower line is comparable to that in Fig. 1 except that it represents

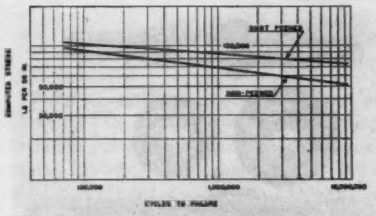

FIG. 2 FATIQUE CHART OF CARBURIZED AUTOMOTIVE-TYPE SPUR AND HELICAL GEARS, SHOTPEENED AND NONSHOTPEENED

carburized gears only and is somewhat higher. The original data in Fig. 1 included carburized gears as well as through-hardened and cyanided gears and, although the test points for the through-hardened gears were somewhat lower than for those which were carburized, there were not sufficient data at that time to distinguish between these two groups. More recent data on carburized gears have indicated that there is a distinct advantage in carburized gears as compared to those which are through-hardened and cyanided. Therefore it would be expected that results on through-hardened gears would fall below the average line. The actual test points have been deleted in Fig. 2 for the sake of clarity.

The upper line in Fig. 2 shows the average life in relation to the calculated stress for shotpeened carburized gears. In all cases, peening was the last operation in so far as the teeth are concerned, and no attempt was made to protect the tooth flanks from the blast.

A chart similar to that in Fig. 2 was recently published by J. O. Almen (6), which shows the lines for carburized gears, peened and nonpeened at a somewhat higher stress. This is due to the fact that the lines in Almen's chart are chosen to represent higher quality of manufacture. The lines shown in Fig. 2 represent a conservative average of the same test data along with additional data accumulated by the author on carburized gears, peened and nonpeened.

It can be seen from the chart that, for example, an average life of 800,000 cycles is obtained on nonpeened gears at a calculated stress of 71,000 psi. For the same average life, the allowable stress for shotpeened gears is 88,000 psi, or an increase of more than 24 percent in allowable stress.

Looking at the chart from the standpoint of increase in life at a given calculated stress, it can be seen, for example, that at a calculated stress of 80,000 psi, the average life of nonpeened gears is 300,000 cycles, and that of shotpeened gears 3,000,000 cycles, or an increase of 1000 percent in life.

Shotpeening is equally effective on gears of lower hardness as well as other types of gears, such as spiral bevel, hypoid, and so forth. Sufficient data are not available for stress-life charts on the other types, but an increase of 10 percent in allowable stress is quite conservative.

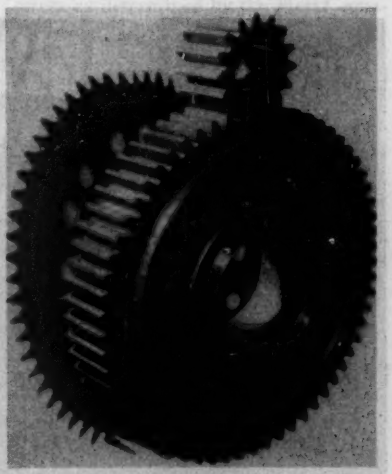

FIG. 3.—Locomotive Drive Gears Shotpeened in Regular Production.
(Courtesy Electro Motive Division, General Motors Corporation)

In most cases, shotpeening is the last operation in so far as the teeth are concerned. The slight roughening of the surface of the tooth flanks has no detrimental effect. Actually some manufacturers feel that shotpeened gears are more quiet-running than nonpeened gears of the same design and manufacture.

In some cases, however, it is desired to have a smooth finish on the tooth flanks. This can be accomplished by cutting the teeth with a protuberance hob, which produces an effect similar to undercutting at the root of the tooth. The gears are then hardened, shotpeened, and ground. This procedure allows the tooth flanks to be ground without removing any of the shotpeened surface in the fillet where the bending stress is maximum.

Fig. 3 shows an example of large gears which are shotpeened in regular production. These gears are used in a railroad application. Fig. 4 is an example of smaller gears used in heavy-duty truck transmissions, also shotpeened in regular production. These are representative of many applications in which shotpeening is considered by the manufacturer as an important part of the production processing.

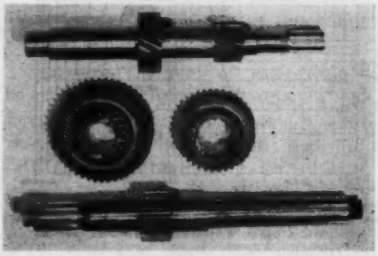

FIG. 4.—Heavy Duty Truck Transmission Gears Shotpeened in Regular Production. *(Courtesy Fuller Manufacturing Company)*

Fig. 5 illustrates a machine used for shotpeening gears in production.

FIG. 5.—Machine for Shotpeening Gears in Production.

PITTING RESISTANCE

Data on pitting resistance are not as plentiful as is the case for bending strength, primarily because of the difficulty in obtaining quantitative data. Bending failures can be recognized easily while the gears are running, because of the sudden noise. Pitting, on the other hand, is a very slow and progressive failure, starting with extremely small craters in the tooth profile. In some cases these craters grow larger and more numerous until finally sufficient material is removed so that complete failure results. In other cases it may develop as very small pits which progress to a moderate degree and then stop without any real damage.

Wickendon, Brophy, and Miller (*7*) have been successful in establishing some quantitative data, and in determining the life of gears at which pitting begins. These tests were run on a special test machine designed to minimize variables in testing. It is to be hoped that such tests will add appreciably to the information now available on pitting. It is rather generally accepted that pitting is the result of high compressive stress, but, to the author's knowledge, there are not sufficient data available to establish the validity of any method of calculation on the basis of actual fatigue tests. A value of maximum compressive stress can be obtained by the method described later in this discussion under Scoring Resistance. A limiting value of 200,000 psi has been used with some success. This is a qualitative value, but it is believed to be on the conservative side.

SCORING RESISTANCE

Scoring is caused by a combination of high compressive stress and high sliding velocity on the contacting tooth surfaces. By means of an approach similar to that just described in appraising the bending-strength formula, a method of calculation of scoring resistance in spur and helical gears was developed by the author, under the direction of J. O. Almen, at the Research Laboratories Division, General Motors Corporation. Dynamometer test data were accumulated on a very large number of gears, along with the complete design information. With these data at hand, various assumptions were made until good correlation was obtained between calculated values of scoring resistance and the actual test data.

Briefly, the method selected (*6*) is based on the same assumption, with regard to distribution of the transmitted load, as that used in the bending-strength calculation discussed in the foregoing. The method consists of the calculation of the product PVT.

Term P is the maximum compressive stress as determined by the Hertz equation for cylindrical surfaces, for a point located at the tip of the gear tooth or pinion tooth, and based upon the total tooth load derived from torque, the average total length of lines of contact, and the curvatures of the tooth surfaces in the plane normal to the line of contact at the selected point. V is the sliding velocity of the surfaces at the selected point. T is the distance in the plane of rotation from the pitch point to the selected point.

The degree of correlation of the calculated values with actual test results can be seen from the chart in Fig. 6 (reference 6).

The data shown in this chart represent actual test results on well over 50,000 pairs of aircraft gears. The gears tested covered a wide range of requirements. The torque requirements varied from a few foot-pounds up to several thousand foot-pounds, and the speed requirements ranged from a few hundred

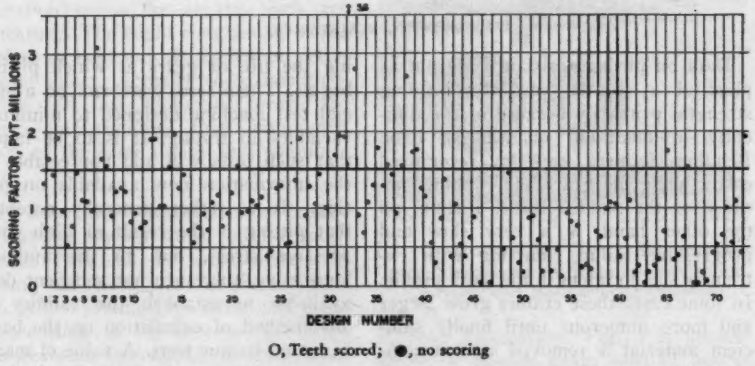

FIG. 6 CORRELATION OF SCORING FACTOR, *PVT*, WITH ACTUAL TESTS ON 73 GEAR DESIGNS
(Pinion tip, left of line; gear tip, right of line; safe limit = 1,500,000 *PVT*.)

rpm up to 28,000 rpm. All of the test data accumulated were on fully hardened spur gears, lubricated with mineral oil. External as well as internal gears are included.

Note that, by and large, those gears which have a calculated *PVT* factor in excess of 1,500,000 failed by scoring, whereas those with a *PVT* factor of less than 1,500,000 had no scoring failure. There are a few exceptions, but not without reason. For example, point No. 6 represents a gear in a high-ratio pair in which, with a true involute profile, tooth action would have taken place very close to the base circle, resulting in a high compressive stress, along with high sliding velocity. The calculated *PVT* factor would indicate scoring failure. However, in this case, the gears were made with a pronounced profile modification at the tip of the gear teeth, thus decreasing the actual compressive stress in that region. It should be mentioned, however, that this particular design was studied and it was found that a *PVT* factor of less than 1,500,000 could have been obtained by decreasing the gear addendum and correspondingly increasing the pinion addendum, without tooth-profile modification. The two points which show scoring failure at a low *PVT* value were in planetary units with a large number of planetary pinions. In both cases, the load was assumed to be distributed equally among the pinions which is probably erroneous. Assuming fewer pinions carrying the load, the *PVT* factor would exceed the safe limit of 1,500,000.

It should be mentioned that with extreme-pressure lubricants, a value of *PVT* in excess of 1,500,000 may be used.

BALANCED DESIGN WITH SHOTPEENING

A study of the design considerations from the standpoint of both bending strength and scoring resistance reveals that there is some conflict between high bending strength on the one hand and high scoring resistance on the other. For example, according to the formulas, a coarse pitch is desirable for bending strength because of the greater thickness at the root of the tooth. However, in some cases this would be impractical because a coarse pitch would necessi-

tate long teeth for continuous action. This, of course, would mean an increase in the length of action and consequently, higher sliding velocity. Furthermore, with long teeth, particularly in high-ratio gears, tooth action approaches the base circle of the pinion which in turn results in a high compressive stress because of the small radius of curvature of the pinion tooth in that region. In such a case, an increase in bending strength may lead to scoring tendencies.

Therefore, in order to obtain satisfactory operation with minimum weight, it may be necessary to compromise between bending strength and scoring resistance. This leads to a logical question as to the usefulness of shotpeening relative to scoring resistance. For a given design, experience indicates that shotpeening has little direct influence on the scoring tendency of gears of the same design and operating conditions. However, since a balanced design may involve a compromise between bending strength and scoring resistance, a distinct advantage can be taken by designing the gears with a finer pitch in favor of scoring resistance. This would result in decreased bending strength which can be restored by shotpeening. By this reasoning it can be seen that shotpeening can be used directly for increasing bending fatigue strength, or indirectly, by proper design consideration, for increasing scoring resistance.

CONCLUSION

It is common experience that certain design factors can be used to improve bending strength. It is also generally accepted that bending strength can be increased materially by shotpeening. However, the possibilities of shotpeening, in combination with appropriate design, as a means of increasing scoring resistance probably has not been appreciated. The intention in the foregoing discussion is to point out some of the advantages of shotpeening in combination with the design of gears as a means of (*a*) reducing bending failures, (*b*) reducing scoring failures, (*c*) reducing weight or space requirements, (*d*) reducing production costs.

With respect to reducing production costs, the extent of such reduction would vary considerably for various applications, but it might be of interest to cite an example of what can be accomplished. An estimate of possible reduction was made on the basis of a production machine used for shotpeening large coil springs. Assuming an increase of 10 percent in allowable stress, the saving in material only, by virtue of decreased size, was estimated at $25 per hr. of machine operation after all peening costs were taken into account. The estimate was very conservative in every respect.

ACKNOWLEDGMENTS

The author is sincerely grateful to Mr. J. O. Almen of the Research Division, General Motors Corporation, and to Mr. A. H. Candee of the Gleason Works for valuable suggestions and helpful criticism during the preparation of this paper.

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PROGRESS IN MARINE ENGINEERING

Past, Present and Future

ACKNOWLEDGMENT

"The British Motor Ship" for July, 1951, contained this article by T. A. CROWE, Engineering Director, John Brown and Co., Ltd. This presentation is the concluding part of a paper read at the 1951 Joint Engineering Conference of the Institutions of Civil Engineers, Mechanical Engineers, and Electrical Engineers.

It is found from an analysis of the figures compiled by Lloyd's Register of Shipping that approximately 63 percent of the tonnage of ships building in Britain at June 30, 1950, were to be propelled by Diesel engines (62 percent at March, 1951).

The modern oil engine is attractive owing to its low fuel consumption of about 0.36-0.37 lb. per b.h.p.-hr., and when it is remembered that most modern engines will operate satisfactorily on boiler oil, it becomes doubly attractive. The internal combustion engine of today also meets the all-important requirements of reliability and low maintenance costs.

With a modern Doxford engine, fuel consumption as low as 0.34 lb. per b.h.p.-hr. have been obtained, corresponding to a thermal efficiency of nearly 40 percent, which makes this engine the most efficient prime mover in use at sea today.

At the time of writing no gas turbines have been fitted in a mercantile vessel, but in 1947 Messrs. Metropolitan-Vickers installed a 2500 s.h.p. open-cycle gas engine in M.G.B. 2009 of the Royal Navy, and in 1950 Messrs. Rover installed 120 s.h.p. gas turbines in a 60-ft. launch. Probably the first mercantile vessel to be fitted with a gas turbine will be the M.S. *Auris*.

The British Thomson-Houston Company have constructed a 1200 b.h.p. open-cycle gas turbine which is intended to replace one of the four Diesel-electric generators. A diagrammatic arrangement of this gas turbine, which will shortly be installed in the ship, is shown in Fig. 2.

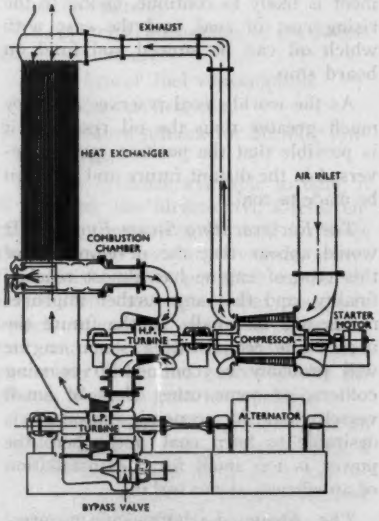


Fig. 2.—Arrangement of B.T.H. 1,200 b.h.p. open-cycle gas turbine.

The fuel consumption at 1200 b.h.p. is estimated to be 0.7 lb. per b.h.p.-hr., corresponding to an over-all thermal efficiency of 20 percent. The designed

temperature of the gases entering the high-pressure turbine is 1200 degrees F.

In 1946 Pametrada designed, and produced working drawings for a 3500 s.h.p., open-cycle, marine gas turbine. The work of constructing the various units involved was allocated among eleven member firms. This gas turbine is at present undergoing shop trials on the Pametrada test bed at Wallsend. A diagrammatic layout of this unit is shown in Fig. 3.

Fuels.—Oil fuel is rapidly replacing coal as the fuel used in all vessels in the mercantile marine. This development is likely to continue, owing to the rising cost of coal, and the ease with which oil can be stowed and fired on board ship.

As the world's coal reserves are very much greater than the oil reserves, it is possible that the position may be reversed in the distant future and a return be made to coal.

The Reciprocating Steam Engine.—It would appear that the development of this type of engine has almost reached finality, and that any further improvements will be small. In the future the use of the reciprocating steam engine will probably be confined to coasting colliers, or some other class of small vessel where, for economic reasons, it is desirable to burn coal, and where the power is too small for the installation of an efficient steam turbine.

The Steam Turbine.—Steam pressures will continue to increase but not in any spectacular manner. In the 1920's the steam pressures in use in turbine vessels were approximately doubled, but this was due to the substitution of the water-tube for the Scotch boiler, which removed the limitation on steam pressures which had existed for so long. No such limitation on the turbine designer exists today and he can choose his steam pressure to suit his turbine. For this reason it is felt that the steam pressures at present recommended by Pametrada

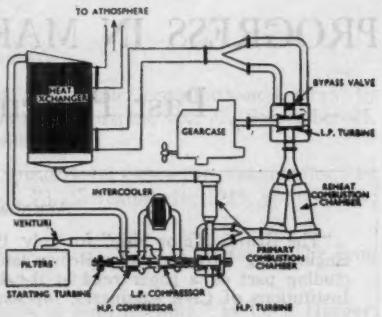


Fig. 3.—Arrangement of Pametrada's 3,500 b.h.p. open-cycle gas turbine.

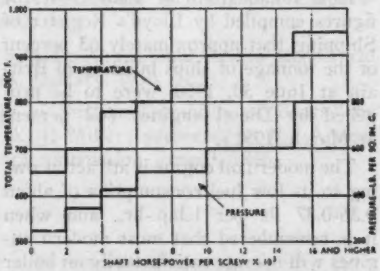


Fig. 1.—Steam conditions recommended by Pametrada, 1947.

(Fig. 1) are not likely to be exceeded by more than perhaps 25 percent for many years to come.

With the development of special steels and other metals having improved creep properties at high temperatures, the use of higher steam temperatures is to be expected.

For steam temperatures of 1050 degrees F., the material of the tubes in the final passes of the superheater would require to be of either austenitic steel or one of the nickel-chrome alloys such as "Nimonic."

When considering the use of steam at these or higher temperatures on board ship, careful consideration will require to be given to the problem of "vanadium attack." All residual oils contain

vanadium to a greater or lesser degree, and when combustion takes place this vanadium combines with oxygen and forms vanadium pentoxide, which at tube temperatures above about 1150-1200 degrees F. causes serious corrosion. Experiments to date indicate that Nimonic tubing is more resistant to vanadium attack than any of the alloy steels.

If steam temperatures of more than 1200 degrees F. are used, trouble may be experienced due to dissociation of the steam into hydrogen and oxygen, the oxygen causing rapid corrosion.

Auxiliaries.—The use of electric auxiliaries will probably become even more general than it is today. Owing to the simplicity, reliability, and low first cost of alternating-current motors, their use at sea will almost certainly increase, until they largely supersede direct-current motors as they have done on land.

In time the steam turbine and Diesel generators will probably be replaced by gas-turbine driven alternators.

The Diesel Engine.—It is difficult to forecast the future developments of the Diesel engine, but it appears safe to say that the four-cycle engine will disappear for main propulsion.

Whether the loop-scavenge, single-piston engine will survive, or whether it will be replaced by either the opposed-piston or some other form of straight-through scavenge is doubtful, but in view of the improved scavenging obtained with the latter type it would appear to be the engine of the future.

Whichever type of two-cycle engine does survive it is likely to be supercharged, probably by an exhaust gas turbine-driven supercharger. It is possible that supercharging will be further increased, and the exhaust gases released from the cylinders at a relatively high pressure into a gas turbine. If this is done, then the gas turbine will develop more power than is required to drive the supercharger, and this excess power can be used to drive the propeller shaft through gearing. In the limit the Diesel engine becomes a "power gas generator" and all the useful work is developed by the gas turbine. Messrs. Sulzer Brothers of Winterthur have carried out a considerable amount of experimental work on this cycle.

The use of geared Diesel engines is gaining ground at the present time and is likely to continue. The principal advantages of the geared engines are:—

(1) Lower head room required.

(2) Ability to obtain an increased power per shaft.

(3) Individual parts are lighter and easier to handle.

The principal advantages of the direct-drive engines are:—

(1) Lower fuel consumption.

(2) Lower first cost.

(3) More suitable for burning low-grade oils.

On the evidence available to date, it is felt that the direct-drive Diesel engine will continue to be used for some time to come for cargo vessels, where the requisite power per shaft can be obtained from one engine. For passenger vessels where low headroom is important, and for vessels requiring a large power per shaft, the geared Diesel engine is likely to gain in popularity.

The Free Piston Compressor.—As an alternative to the Diesel-engine cycle, a Pescara free piston compressor can be used to supply hot, pressured exhaust gas to the gas turbine.

The fuel consumption with either the "power gas generator" or with the free-piston compressor would be about the same as it is with the normal Diesel engine.

The Gas Turbine.—In the last decade the gas turbine has made rapid advances in the air and on land, and it is only a matter of time before it is used in the mercantile marine. The design of a gas turbine for a merchant ship would be based more on that of industrial gas turbines than on aircraft gas turbines.

There are at present building in Britain three large gas turbines for driving alternators.

In order to be competitive with the Diesel engine, a gas turbine must be capable of burning boiler oil, and must have a fuel consumption of not more than 0.4 lb. per s.h.p.-hr. To realize this consumption it will be necessary to increase the initial temperature of the gas or air entering the turbine to approximately 1350 degrees F. or about 100-150 degrees F. above that contemplated today for long-life industrial gas turbines.

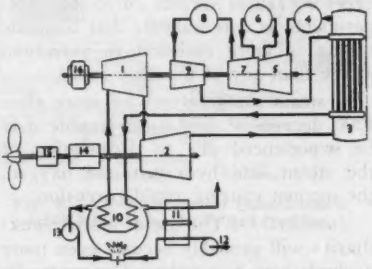
When it is compared with the steam turbine, the gas turbine shows up to better advantage. With present-day materials, a closed-cycle gas turbine could be designed for a mercantile vessel of 10,000-15,000 s.h.p. per shaft, to have a fuel consumption of approximately 0.45 lb. per s.h.p.-hr.

Fig. 4a shows a schematic diagram of a 12,000 s.h.p. closed-cycle gas turbine, and Fig. 4b shows a proposed layout of this machinery in a 9500-ton cargo ship.

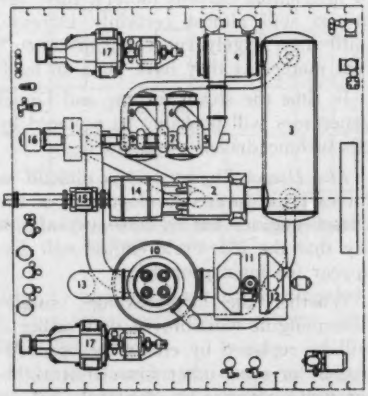
The Diesel engine will probably hold its own, for powers up to 5000-6000 b.h.p. per shaft, for many years to come, but no reason is seen why, when the gas turbine has demonstrated its reliability for base-load power generation ashore, it should not be adopted for marine use, particularly for the larger powers. Ultimately, it is felt that the gas turbine will completely displace the Diesel engine and the steam turbine for the propulsion of ocean-going mercantile vessels.

Nuclear Energy.—It may be of interest to consider briefly the possibility of the use of nuclear energy for the propulsion of ships.

It has been reported that the application of nuclear energy to the propulsion of ships is to be studied at Pocatello, in Idaho, where an experimental reactor is to be built by the Westinghouse Electric Corporation as a prototype for a marine unit. The cost of this plant is estimated to be 25-30 million dollars and it will run on uranium metal enriched with U235.



(a) Gas turbine cycle.



(b) Layout of machinery in ship.

Fig. 4.—12,000 s.h.p. closed-cycle gas-turbine installation.

1.—High-pressure turbine. 2.—Low-pressure turbine. 3.—Heat exchanger. 4.—Precooler. 5.—Low-pressure compressor. 6.—Intercooler. 7.—Intermediate-pressure compressor. 8.—Intercooler. 9.—High-pressure compressor. 10.—Air heater. 11.—Combustion-air preheater. 12.—Forced-draught fan. 13.—Recirculating gas fan. 14.—Reduction gears. 15.—Thrust block. 16.—Starting motor—auxiliary generator. 17.—Auxiliary gas turbo-generators.

A schematic diagram of a nuclear-energy gas-turbine cycle is shown in Fig. 5. In this cycle, heat is generated by the fission of uranium in a nuclear reactor or pile, and this heat is removed by passing some suitable fluid through

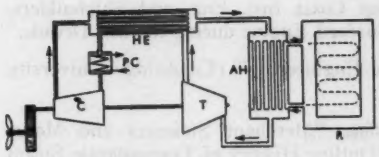


Fig. 5.—Diagram of nuclear-energy gas-turbine cycle.

AH.—Air heater. C.—Compressor. HE.—Heat exchanger. PC.—Precooler. R.—Reactor. T.—Turbine.

the reactor. This fluid, which is probably radio-active, is passed through an air heater where it gives up heat to the air in a closed-cycle gas turbine circuit. This hot air, which is also at a high pressure, is expanded in a turbine where it generates power. After leaving the turbine the air passes through a heat exchanger and a water-cooled precooler before entering the air compressor. The compressed air leaving the compressor is heated in the heat exchanger before again entering the air heater. Part of the power generated by the turbine is used to drive the compressor, and the remainder is available for driving the propeller.

The fuel consumption of a nuclear reactor has been calculated to amount to about 1 gram of uranium per day for an output of 1000 kw. On this basis the consumption of uranium required to propel the *Queen Elizabeth* from Southampton to New York would be 1¼ lb. But when it is remembered that the cost of uranium is about £125,000 per lb. the picture does not look so bright.

In the reactors which have been constructed up to the present it has only been possible to remove the heat at a temperature of about 550 degrees F., because the aluminum containers for the fuel rods, in which most of the heat is produced, deteriorate rapidly at high temperatures. The efficiency attainable in converting this low-grade heat into useful work is necessarily low, and means will have to be found to raise this temperature very considerably before a reasonable efficiency can be obtained.

Another serious problem to be overcome before nuclear energy can be successfully used for power generation at sea is the necessity to shield the reactors to prevent the escape of neutrons and radio-active rays. This shielding would require to consist of not less than 6 ft. of concrete, or a similar weight of another material.

During the fission chain reaction, other products are formed, some of which remain radio-active for years, and their disposal presents a serious problem. The small quantities which have been produced up to now have been concentrated by chemical means and either buried deep underground or sealed in metal containers and dropped into the sea.

From a consideration of all the above questions it is felt that nuclear energy is not likely to be used for the propulsion of mercantile vessels for many years to come.

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BOOK REVIEWS

AIR TRANSPORTATION MANAGEMENT, Its Practices and Policies by Joseph L. Nicholson. Published by John Wiley and Sons, Inc., 440 Fourth Avenue, New York 16, N. Y. 446 pages. Price $6.50.

Mr. Nicholson's treatment of the subject is timely in that it stresses the part which is played in industry by the government. The economic aspects of day-to-day operations and of long-term profits and losses rather than history and descriptions predominate.

Each chapter in the book is a complete unit and each has been read by at least four experts in the field which is covered.

MARINE NEWS 1951 DIRECTORY. Twenty-seventh Annual Edition. Published annually by Marine News, 26 Water St., New York 4, N. Y.

As usual the Directory presents a thorough listing of facts and persons which are of interest or importance to those who operate in the maritime field.

The Society has the sad duty of announcing that it has been informed of the death of the following members since the publication of the November, 1951, Journal:

LANDBERG, ERIK G., Civil Member

LUCKENBACH, J. LEWIS, Civil Member

ORR, FRANK L., Associate Member

It is with much pleasure that the Society announces that the following have joined since the publication of the November, 1951, JOURNAL:

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Boddy, R. R.
Folsom, Frank M.
Forde, Ivan
Knight, A. Raymond

DROPPED

Pursuant to provisions of the By-Laws, it was necessary to drop from our rolls 165 members who had become 2 years in arrears for dues.

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $40.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

Subject Matter and Authors

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(a) What subject would you, as a member, like to see covered by an article in the Journal.

and

(b) Who do you consider best qualified to prepare such an article authoritatively, and what is his address (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

PHOTOGRAPHS

Photographs of current or historical interest to readers of the Journal are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

LIFE MEMBERSHIPS

Any member may now purchase a life membership under the following rules:

(a) Any dues paid for the year in which life membership is purchased will be credited to the cost of the life membership.

(b) Life memberships are non-transferable and terminate with the death of the member.

(c) No refund will be made on account of death or resignation of a life member.

(d) Life membership will vary in cost with the age of the member at his next birthday following application, as follows:

Over 55 years	$125.00
40 to 55 years	175.00
25 to 39 years	225.00
Under 25 years	250.00

(e) All life memberships paid in shall be protected by a transfer on the books of the Society of Series G, U. S. Government 2½ per cent bonds. Thus a life membership will be established. It will be credited with moneys paid in for life memberships and debited with the regular amount of annual dues for each life member each year.

LOCAL CHAPTERS

Local Chapters of the American Society of Naval Engineers may now be authorized by the Council. The following rules for recognition of Local Chapters have been adopted by the Council:

(a) Twenty or more regular members, Naval or Civilian, may apply for a Charter as a Local Chapter.

(b) Each application must be accompanied by a list of members and a copy of local By-laws. The latter must meet the following minimum requirements.

1. All financial dealings of the local Chapter must be independent of the Society and can in no way obligate the Society as a whole. The local chapter shall set up on a non-profit basis. All accounting shall be local.

2. Each active local Chapter shall receive from the Society a grant of $0.50 per year for each naval member, civil member or associate who resides in the local area and who associates himself with the local Chapter. This grant may be used by the local Chapter to defray any necessary expenses, including (a) Letterheads, (b) Notice cards, (c) Postage, (d) Minimum rental allowance, (e) Secretarial service (part time).

3. Any technical papers presented at local chapter meetings, shall be submitted to the Secretary-Treasurer of the Society for consideration for publication in the JOURNAL. Any paper accepted will be paid for at regular rates.

4. Each local chapter shall submit a quarterly report of its activities to the Secretary-Treasurer. These reports must reach Washington, D. C. prior to 15 January, 15 April, 15 July and 15 October. This report shall contain a list of active local members as of the first day of the quarter for the purpose of computing the amount of grant.

5. Associate members of the Society shall be eligible for membership in a Local Chapter, entitled to all the privileges of other members except voting and holding office.

Permission is granted to reprint any original article contained herein if the following conditions are met:

a) Credit is given to the JOURNAL with reference to the issue.

b) Credit is given to the author.

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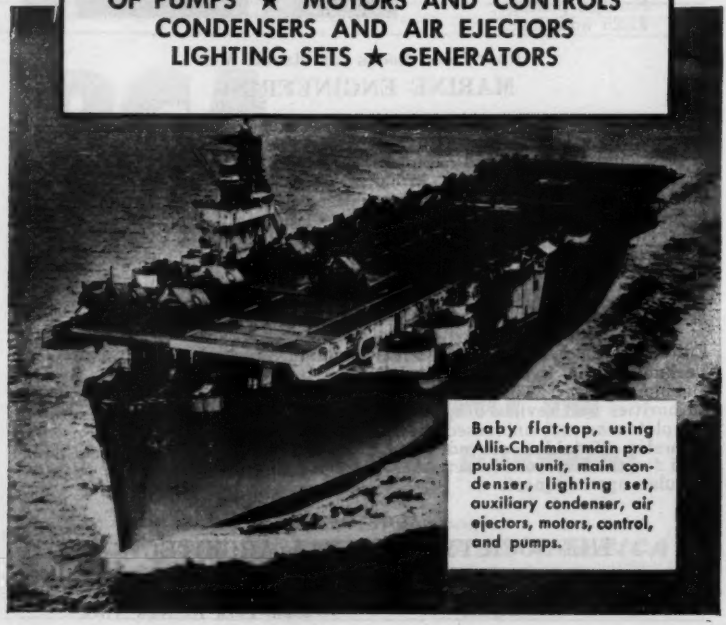

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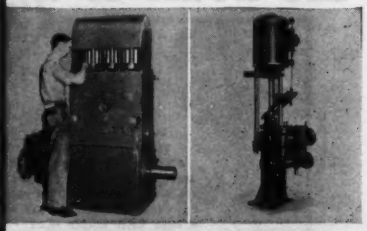

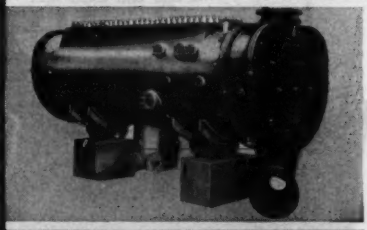

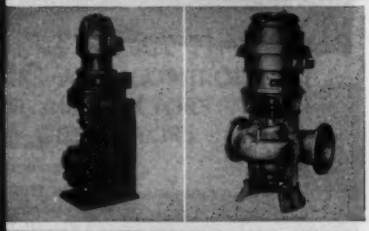

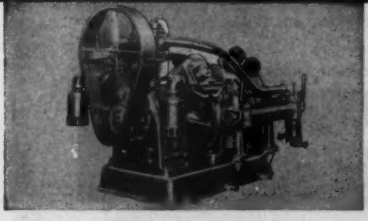

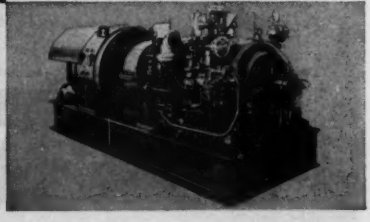

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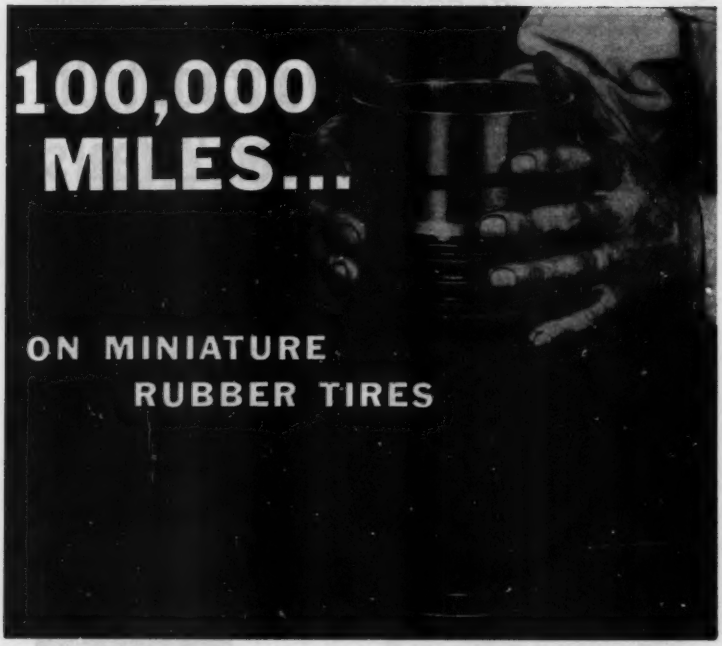

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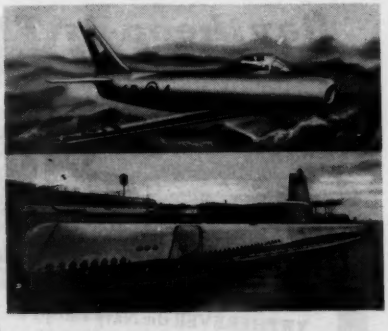

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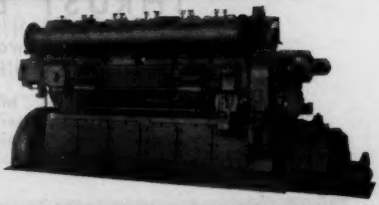

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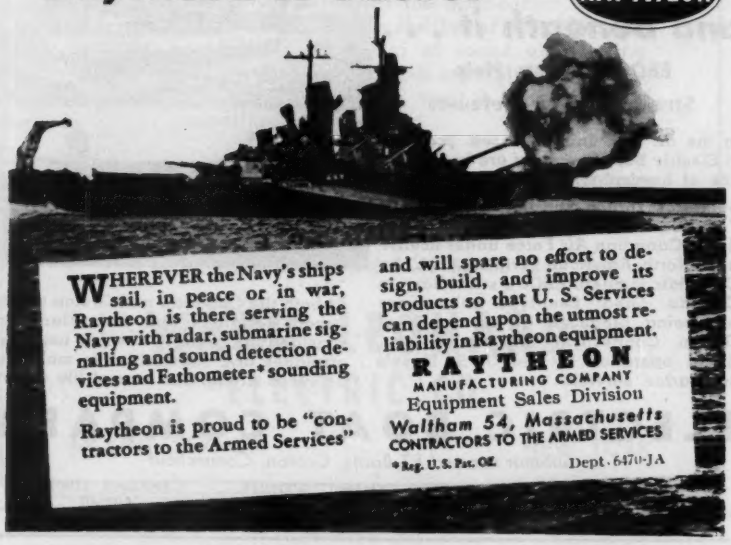

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Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as civil members. They shall have been in the active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

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Name _____

Years in engineering work _____

Years in responsible charge of important work _____

Present business connection and position _____

Recommended by (two members) _____

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For Associate Membership

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Name _____

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Business connection and position _____

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Signature of Applicant

Address for Journal and Mail

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*See reverse side for required qualifications for various classes of membership

JOURNAL OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.
FEBRUARY, 1952.
PART II

BY-LAWS OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

2. The object of the Society shall be to promote a knowledge of marine engineering and Naval architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all parts of the world and publishing them in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed of value to the advancement of the science; and by recording historical events in the lives of engineers. The Society shall not be conducted for profit. Should any profit accrue as a result of its operations, such profit shall not be distributed but shall be held for the objects of the Society as set forth herein.

3. The officers of the Society shall be a President, a Secretary-Treasurer, and a Council.

4. The President and the Secretary-Treasurer shall be Commissioned Officers of the regular Navy, Marine Corps, or the Coast Guard of the United States.

5. The Council shall be composed of eleven members, two of whom shall be the President and the Secretary-Treasurer of the Society. Of the remaining members, four shall be officers

of the regular United States Navy; one shall be an officer of the United States Coast Guard; two shall be officers of the United States Naval Reserve or United States Coast Guard Reserve, and two shall be civilians. The President and Secretary-Treasurer shall act as President and Secretary of the Council.

6. Residence in the City of Washington, D. C., or in the immediate vicinity thereof, during the term of office, shall be a necessary qualification for any officer except a Civilian member of the Council.

7. Removal from the city or its immediate vicinity shall constitute a disability on account of which the Council shall declare any office, except that of Civilian Member of the Council, vacant.

8. The President and Secretary-Treasurer shall be elected annually. Other members of the Council shall be elected for a term of two years, one-half being elected each year. All officers shall assume office on 1 January.

9. The President shall exercise the usual duties of that office.

10. The Secretary-Treasurer shall conduct the correspondence of the Society and its financial transactions. Under the supervision of the Council, he shall edit and publish the JOURNAL of the Society. He shall deposit the funds of the Society in its name in a bank approved by the Council and shall withdraw same by check. He shall keep the securities of the Society in a safe deposit box held in its name. He shall submit an annual financial statement which shall be audited by three members of the Society. He shall receive such salary as the Council may provide.

11. The Council may appoint a naval member as Assistant Secretary-Treasurer, who shall perform the duties of the Secretary-Treasurer during the absence of the latter and receive such salary as the Council may provide. This appointment shall be for a stated term which shall not extend beyond the term of office of the incumbent Secretary-Treasurer.

12. The Council shall direct and manage the affairs of the Society and shall supervise the editing and publishing of the JOURNAL. It shall require that all personnel of the Society handling its funds or securities shall be bonded. It shall fill all vacancies which may occur during a year. In case of death or disability of the President, Secretary-Treasurer or a member of the Council, the Council shall declare the office vacant. The Council shall have authority to decide upon questions of disability. The Council shall have authority to employ such clerical assistance as may be required at such rate of compensation as it may deem proper. The Council may authorize the acceptance of subscriptions for the JOURNAL of the Society from those who are not members as hereinafter provided, and also the sale of single copies of the JOURNAL, and of reprints of separate articles, and may fix the prices thereof; it may also authorize the use of certain portions of the JOURNAL for appropriate advertisements, and fix the rate of payment therefor. The Council may authorize the formation of local chapters of the Society under such rules and regulations as it shall adopt. No local chapter shall be empowered to obligate the Society as a whole or any member of the local chapter in any regard whatever.

13. The Council shall meet at the call of the President, and seven shall constitute a quorum for business. Also any two members of the Council may call a meeting thereof.

QUALIFICATION FOR MEMBERSHIP.

14. The Society shall be composed of Naval Members, Civil Members, Associates and Honorary Members.

15. Commissioned and ex-commissioned officers of the regular Navy, Marine Corps and Coast Guard of the United States; warrant and ex-warrant officers of the regular Navy, Coast Guard and Marine Corps of the United States; reserve commissioned and warrant officers of the Navy, Coast Guard and Marine Corps of the United States shall be eligible as Naval Members. Persons eligible as Naval Members shall be admitted upon application and payment of annual dues.

16. Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as Civil Members. They shall have been in the active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

17. Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as Associate Members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

18. Associate members shall be entitled to all the privileges of other members except voting and holding office.

19. The Secretary of the Navy, the Under Secretary of the Navy, the Assistant Secretary of the Navy, the Assistant Secretary of the Navy for Air, the Chiefs of all material bureaus—that is the Chief of the Bureau of Ships, the Chief of the Bureau

of Ordnance, the Chief of the Bureau of Yards and Docks, the Chief of the Bureau of Aeronautics,—and the Engineer-in-Chief of the United States Coast Guard, all ex-Presidents of the Society, all prize essayists, the Deputy Chief of the Bureau of Ships, the Chief of Naval Material, the Chief of Naval Research, and such other persons as the Society may elect, shall be Honorary Members. To be eligible for Honorary Membership, such other person must receive the unanimous vote of the Council before his name may be presented to the Society; and the favorable vote of two-thirds of the members voting shall be necessary for election. The procedure in voting for persons whose names have been presented by the Council as candidates for Honorary Membership shall be similar to that followed for the election of officers. Honorary Members shall be exempt from the payment of dues.

20. The annual dues shall be $7.50, payable on 1 January, in advance. If a member shall be in arrears for a period of two years, the Secretary shall send notice to that effect under date of 1 October of the second year, by registered mail to such member's last known address. If such member's dues shall remain unpaid on the 31st day of December following this notice, his name shall be dropped from the rolls of the Society; Provided, that he may be reinstated by the Council on written application and payment of all arrears. Naval, Civil and Associate Members may be constituted Life Members under such rules and regulations and at such cost as the Council may establish.

21. All moneys, all correspondence and all matter intended for publication shall be sent to the Secretary-Treasurer.

22. In deciding matters pertaining to the Society, none but Naval, Honorary and Civil Members shall be entitled to vote. To constitute a quorum for business a total of at least twelve Naval, Honorary and Civil Members must be present.

23. The Society shall meet annually in the City of Washington, D. C., on the first Tuesday in October, for the purpose of

making and receiving nominations to offices for the following year, and for the transaction of such other business as may be presented. At least one month prior to the annual meeting, the President shall appoint a Nominating Committee, the duty of which shall be to present to the annual meeting at least two nominations for each office to be filled, except as noted below, having due regard to the requirements of Articles 4 and 5 of the By-Laws. In the case of nominations for President of the Society, only one nomination need be presented. Additional nominations, either oral or in writing, may be made at the annual meeting by any Naval, Civil or Honorary Member. At the annual meeting, nominations for each office shall be closed upon a majority vote of the members present. The consent of a nominee to accept office if elected must be assured. Any member may be placed in nomination for any number of offices for which he is eligible.

24. Immediately following the meeting for the nomination of officers, the Secretary-Treasurer shall prepare and mail to each Naval, Honorary and Civil Member, a ballot showing the name of each candidate and the office to which he is nominated. With such ballot shall be mailed two envelopes, one designed to fit inside the other. The smaller envelope shall be marked "Ballot Envelope." The large envelope shall be addressed to the Society and shall be marked on the address side "Vote—Officers of the Society." A space shall be provided on the back of this envelope for the signature and address of a member and the date of voting. Each member shall place an X opposite the name of each candidate for whom he desires to vote. He shall seal his ballot in the smaller envelope and enclose it in the larger envelope. He shall place his name, address and date of voting on the back of the large envelope. The complete ballot shall be mailed so as to arrive at the office of the Secretary-Treasurer prior to December 26th. Voting by proxy is prohibited.

25. Some time between December 26th and December 30th, a committee of three Naval Members appointed by the President shall open and count the ballots. After completion of this count,

a meeting of the Council shall be called to examine the report of this committee. The Council shall then announce the names of the officers elected for the following year. In case of a tie, the presiding officer shall cast the deciding vote. In the event of a candidate receiving the greatest number of votes for more than one office, he shall decide which he will accept.

26. The Council may call a meeting of the Society whenever it deems it necessary or advisable.

27. Motions to amend these By-Laws, or to make new By-Laws, must be in writing, and must lie over at least three months, during which time notice thereof shall be sent to all Naval, Honorary and Civil Members. The assent of two-thirds of the members voting shall be necessary to amend or adopt. The procedure in voting on changes in the By-Laws shall be similar to that followed in the election of officers.

SECRETARIES OF THE SOCIETY

Current:

Captain J. E. HAMILTON, U. S. Navy, Retired

Past Secretaries:

1889	P. A. Engineer R. S. GRIFFIN, U. S. Navy
1890	Assistant Engineer W. M. MCFARLAND, U. S. Navy
1891	Assistant Engineer EMIL THEISS, U. S. Navy
1892-93	P. A. Engineer W. M. MCFARLAND, U. S. Navy
1894-95	P. A. Engineer R. S. GRIFFIN, U. S. Navy
1896-97	P. A. Engineer F. C. BIEG, U. S. Navy
1898	P. A. Engineer W. M. MCFARLAND, U. S. Navy
1899	Chief Engineer A. B. WILLITS, U. S. Navy
1900	Lt. Comdr. A. B. WILLITS, U. S. Navy
1901	Lieutenant B. C. BRYAN, U. S. Navy
1902	Lieutenant C. W. DYSON, U. S. Navy
1903	Lt. Comdr. JOHN R. EDWARDS, U. S. Navy
1904	Lieutenant M. E. REED, U. S. Navy
1905	Lieutenant W. W. WHITE, U. S. Navy
1906	Lieutenant C. K. MALLORY, U. S. Navy
1907-08	Lt. Comdr. T. C. FENTON, U. S. Navy
1909-10	Lieutenant H. C. DINGER, U. S. Navy
1911	Commander U. T. HOLMES, U. S. Navy
1912	Lieutenant JOHN HALLIGAN, U. S. Navy Lt. Comdr. E. L. BENNETT, U. S. Navy
1913	Lieutenant O. L. COX, U. S. Navy
1914	Lt. Comdr. H. C. DINGER, U. S. Navy
1915-16	Lieutenant A. T. CHURCH, U. S. Navy
1917	Lt. Comdr. J. O. RICHARDSON, U. S. Navy Lt. Comdr. F. W. STERLING, U. S. Navy, Retired
1918	Lt. Comdr. F. W. STERLING, U. S. Navy, Retired
1919	Lt. Comdr. F. W. STERLING, U. S. Navy, Retired Commander J. S. EVANS, U. S. Navy
1920	Commander J. S. EVANS, U. S. Navy
1921	Commander J. S. EVANS, U. S. Navy Commander S. M. ROBINSON, U. S. Navy
1922-23	Commander S. M. ROBINSON, U. S. Navy
1924-25	Commander BRYSON BRUCE, U. S. Navy
1926	Commander A. M. CHARLTON, U. S. Navy
1927	Commander H. B. HIRD, U. S. Navy
1928	Commander H. B. HIRD, U. S. Navy Captain O. L. COX, U. S. Navy
1929-30	Commander H. T. SMITH, U. S. Navy
1931	Captain O. L. COX, U. S. Navy
1932	Commander H. F. D. DAVIS, U. S. Navy
1933-34	Commander H. B. HIRD, U. S. Navy
1935	Commander C. S. GILLETTE, U. S. Navy
1936	Commander C. S. GILLETTE, U. S. Navy Commander ROGER W. PAINE, U. S. Navy
1937	Commander ROGER W. PAINE, U. S. Navy
1938	Commander ROGER W. PAINE, U. S. Navy Lt. Comdr. GUY CHADWICK, U. S. Navy
1939-40	Lt. Comdr. GUY CHADWICK, U. S. Navy
1940-44	Captain J. E. HAMILTON, U. S. Navy
1945	Commander R. T. SUTHERLAND, JR., U. S. Navy
1945-48	Captain F. W. WALTON, U. S. Navy